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Form 6

NOTES ON

HEAT AND STEAM

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Preface to Third Edition.

In preparing this edition for the press, many of the chapters have been re-written and considerable new matter has been added.

The chapter on fuels has been made more complete and the subject of chimneys receives special consideration.

The principal addition has been the chapter on gas engines and refrigerating machines.

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It is hoped that the absence of "padding" in this little treatise may commend itself to those who seek information.

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Chapter I.

MEASUREMENT OF HEAT.

1. Heat is the energy of a body due to its molecular motion.

Heat being one form of *energy* or capacity for doing work is therefore subject to the well known law of conservation of energy.

Work done on a body, as in compressing it, increases the molecular velocity and is said to heat the body. Similarly electrical, chemical or light energy may be transformed into heat and vice versa.

2. **Temperature**:—Temperature is a measure of the intensity of heat in a substance but not of its quantity and is analogous to potential in electric energy. Quantity or current is measured by entropy. (See Chapter 4.)

Two bodies are said to be at the same temperature, when there is no tendency for heat to pass from either body to the other.

The two standard temperatures are: 1—That of melting ice at the average atmospheric pressure, or 32° Fahrenheit. 2—That of steam at the same pressure, or 212° Fahrenheit.

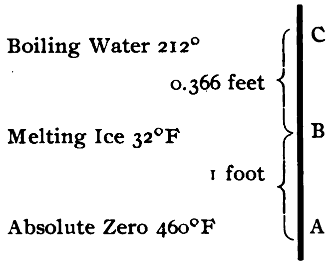
(Hereafter the abbreviation F. will be used for Fahrenheit.)

3. **Pressure**:—The average pressure of the atmosphere is taken to be:—

14.7 pounds per square inch.
2116.3 " " " foot.
29.922 inches of mercury.
33.9 feet of water.

To reduce inches of mercury to pounds per square inch, divide by 2.036.

4. Air Thermometer:—Degrees of temperature are best measured by the expansion and contraction of a perfect gas. Dry air is near enough to a perfect gas for ordinary purposes.



In the diagram, let the vertical line A B C represent a tube containing one foot of dry air at the temperature of melting ice, and enclosed by an air tight movable piston under a constant pressure.

Then it can be shown by experiment that if the temperature rise to that of boiling water, the air will expand until it occupies 1.366 feet of the tube, i. e. it expands 0.366 feet for 180° rise in temperature.

If B C represents 180°, then will A B represent to the same scale:—

$$\frac{180}{.366} = 492^\circ$$

5. Absolute Zero:—The point A or the bottom of the tube is taken as the absolute zero of the air thermometer, and is accordingly 492° below 32 F. or 460° below the Fahrenheit zero. If the air followed this same law below 32° F. indefinitely, then its volume would vanish at —460° F.

The lowest temperature yet observed has been reached by the solidification of hydrogen and is approximately —433° F. or 27° absolute.

6. Absolute Temperature:—From the above it follows that the absolute temperature of a substance is found by adding 460° to its temperature Fahrenheit.

The exact location of the absolute zero for a perfect gas has been computed to be —460.66° F.

Hereafter the symbol t will be used for temperature Fahrenheit, and T for absolute temperature.

7. Change of Pressure and Volume:—Both the pressure and volume of a gas may change with the temperature. Let p, v and T represent respectively the pressure, volume and absolute temperature of a perfect gas; then can it be shown by experiment that within observed limits

pv is proportional to T,

$$\text{or } \frac{pv}{T} = \frac{p_1v_1}{T_1} \text{ a constant.}$$

Let p_1v_1 be the product of the pressure and volume of a gas at 212°F, and p_0v_0 the same at 32°F, then it follows from what goes before, that :—

$$\frac{p_1v_1}{p_0v_0} = \frac{T_1}{T_0} = 1.366$$

8. Quantities of Heat:—Quantities of heat are measured by means of the various changes in bodies which accompany the transfer of heat, as change of temperature, change of volume, melting and evaporation.

Equal changes of temperature do not necessarily imply equal quantities of heat transferred; for instance the quantity of heat required to raise a pound of water one degree will raise a pound of iron about nine degrees.

9. Unit of Heat:—The British thermal unit is the quantity of heat necessary to raise one pound of distilled water one degree Fahrenheit from 39° F, its temperature of greatest density.

This unit will be denoted by the symbol t. u. in these notes.

10. Specific Heat:—The specific heat of water at 39° F is taken as unity; the specific heat of any other substance may therefore be expressed as the fraction of a thermal unit required to raise one pound of that substance one degree Fahrenheit.

The specific heats of nearly all substances are less than unity.

11. Latent Heat:—Latent heat is heat which produces some other change in the substance than that of temperature.

There are latent heats of expansion, of fusion and of evaporation. In each of these the heat which is given to the body produces some molecular change.

For instance, to raise the temperature of a pound of air in a closed vessel one degree Fahrenheit, will require 0.169 t. u.; but to raise the temperature the same amount, when the air is allowed to expand at a constant pressure, will require 0.238 t. u. The difference 0.069 t. u. is the latent heat of expansion of air.

12. Heating of Water:—If heat be continually applied to a pound of ice, the results will be as follows:

From original temperature to 32° F the specific heat will be 0.504. The latent heat of fusion, at 32° will be 142 t. u. nearly, the temperature remaining the same until the ice is all melted.

From 32° to 212° F the specific heat will be nearly unity, being slightly greater near 212°. The latent heat of evaporation at 212° will be 966 t. u., the temperature remaining constant until water is all evaporated.

Above 212° the specific heat will vary from 0.5 to 0.7.

The accompanying diagram Fig. 1, from "STEAM," illustrates the behavior of different substances under the action of heat.

13. Transfer of Heat:—The rate of transfer of heat between two bodies at different temperatures depends:—

1.—On the two temperatures and their difference.
2.—On the area of the surface of contact.
3.—On the nature of the materials.
4.—On the nature and thickness of any intervening substance.

The rate of conduction of heat may be expressed in thermal units per square foot of area per hour.

Let $q =$ the quantity of heat so expressed.
$x =$ thickness of intervening plate in inches.
t_1 and $t_2 =$ temperatures on either side of plate.

Then $q = \frac{t_1 - t_2}{rx}$ where r may be called the co-efficient of internal resistance of the intervening substance.

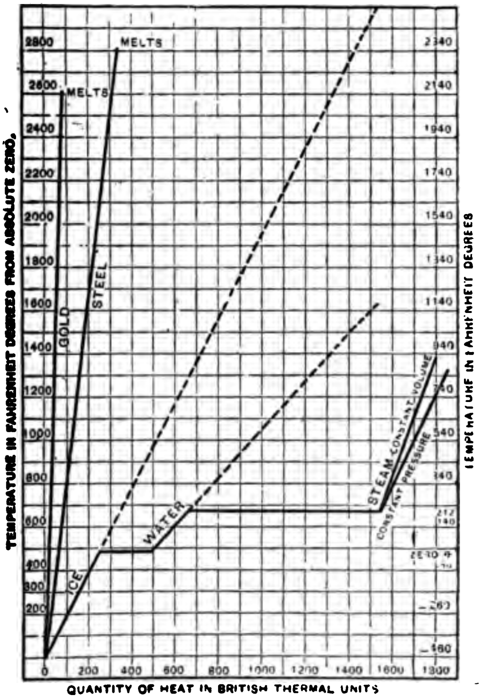


Figure 1

The following table of values of r is given on the authority of Rankine:

TABLE I.—Co-efficients of Thermal Resistance.

Gold, platinum, silver	.0016
Copper	.0018
Iron	.0043

Zinc	.0045
Lead	.0090
Marble	.0716
Brick	.1500

When a metal plate has a liquid on one side of it and a hot gas on the other, as in the case of a boiler plate, there is also an external resistance to conduction, which depends in a marked degree on the nature and condition of the surface. A rough formula deduced from experiments on steam boilers is:—

$$q = \frac{(t_1 - t_2)^2}{a}$$

where a varies from 160 to 200, and q is independent of x. The great difference between these two formulas for transfer of heat is due to the fact that the first formula takes no account of the resistance of the surfaces, which in the case of a boiler shell is the principal resistance. A vertical surface will transmit only about two-thirds as much as a horizontal one.

Grease offers about 1000 times and boiler scale about 100 times the resistance of steel.

The resistance due to the soot and scale on a boiler plate may be graphically illustrated by Fig. 2, where the three layers are as named and the oblique lines show the relative drops in temperature as the heat passes from fire to water.

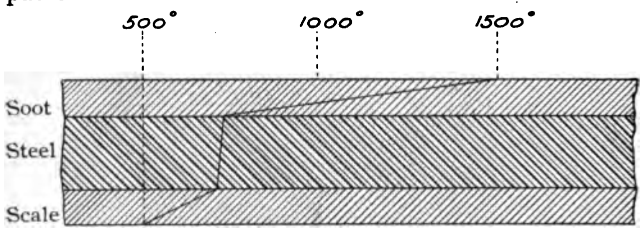


Figure 2

Experiments on the steam boilers at the Centennial Exposition showed, that one square foot of heating

surface transferred heat enough to evaporate three pounds of water per hour under average conditions. The results of a large number of tests of stationary boilers by Barrus show an evaporation of from 1.3 to 6.2 lbs. water per square foot of heating surface with an average of about 3 lbs.

The transfer of heat from a radiator or coils to the air of a room may be expressed in several different ways. Prof. Hutton states that with the direct radiation system the transfer will amount to 2 or 3 t. u. per square foot of surface per degree difference of temperature per hour.

He further states that where cool air is blown over hot coils as in the indirect system one square foot of surface with steam at 212° F will heat 100 cu. ft. of air from zero to 150° F per hour or 300 cu. ft. from zero to 100° F per hour.

A feed water heater or a surface condenser is another example of the transfer of heat. Experiments have shown the transfer in such cases to be independent of the thickness of the tube, within practical limits, and to be proportional to the difference of temperature. Prof. Whitham gives a formula for such cases, as follows:

$$S = \frac{W\ L}{180\ (T_1 - t)}$$

Where S = surface of tubes in square feet.
W = weight of steam condensed per hour.
L = latent heat of saturated steam at temp. T_1.
T_1 = temperature of steam.
t = mean temperature of water.

The formula is for brass tubes:

The usual allowance in feed water heaters is 1-3 sq. ft. heating surface per 30 lbs. of water heated per hour, the latter being a nominal horse-power for boilers.

EXAMPLES.

1.—If a piece of wrought iron weiging 8 lbs. and

having a temperature of 1800° F be plunged into a barrel containing 184 lbs. of water at a temperature of 60° F, what will be the resulting temperature, the specific heat of wrought iron being .1138?

2.—Determine the total number of thermal units required to convert a pound of ice at 0° F into steam at 212° F.

3.—The temperature of the fire under a boiler is 1200° F and the temperature of the water is 212° F, the plates being of iron 3/8 inches thick. Determine the amount of water evaporated per square foot of surface per hour.

(1)—Neglecting the external resistance of the plate.

(2)—Using the approximate formula for boilers on page 7.

4.—A certain condensing engine of 500 horse-power uses 16 lbs. of steam per horse-power hour. Determine the number of square feet of tubes needed for the condenser if the temperature of the steam is 170° F and the initial and final temperature of the water 54° and 142°.

5.—How many square feet of heating surface should be allowed per nominal horse-power of a boiler?

EXAMINATION.

1.—Explain and illustrate conservation of heat energy.

2.—Why are the temperatures of steam at atmospheric pressure and of melting ice used as standards? Why not use boiling or freezing water?

3.—What is meant by the Absolute Zero of temperature and is it a natural or an artificial zero?

4.—Illustrate by a diagram the law of change of pressure and volume of a perfect gas and show how this locates absolute zero.

5.—Define Thermal Unit and Specific Heat in your own language.

6.—Criticise the term Latent Heat.

7.—Explain the discrepancy between the two formulas for the transfer of heat through boiler plate.

Chapter 2.

COMBUSTION AND FUEL.

14. **Combustion**:—is a rapid chemical combination, accompanied by the production of heat and is distinguished from ordinary oxidation or from explosion by the degree of rapidity with which the combination is effected.

Practically it is the combination of carbon or hydrogen in the fuel with the oxygen in the air supplied by the draft.

15. **Fuel**:—A fuel is then any substance occurring in nature which contains carbon or hydrogen in such form or such proportions as to burn steadily and quietly. The principal fuels in the order of their geologic ages are: wood, peat, lignite, bituminous coal, petroleum, natural gas, semi-bituminous coal, anthracite coal.

Briefly—Wood is vegetable matter containing a small amount of carbon and a large amount of water and ash. Peat is derived from mosses and grasses which have been subjected to pressure and heat and when dry usually contains more carbon than does wood. Lignite is an imperfectly formed coal of a brownish color and frequently showing the grain of the wood from which it was formed, as peat was formed from less solid material. Bituminous coal is further advanced in its formation than lignite, contains more carbon and available hydrogen and less waste material. It is accordingly a better fuel.

Petroleum and gas are hydro-carbons in a liquid and a gaseous state respectively, and have a high fuel value as there is practically no incombustible material present. Anthracite is practically pure carbon and is more difficult to ignite than bituminous coal on account of the absence of hydrogen.

The so-called semi-anthracite and semi-bituminous coals, as their names imply, are intermediate between the hard and the soft coals and contain a small amount of hydrogen.

The principal artificial fuels are coke and illuminating gas, both formed from bituminous coal by a process of distillation. Coke is nearly pure carbon and illuminating gas a hydro-carbon but usually inferior to natural gas.

Producer-gas is a product of the complete destructive distillation of bituminous coal and contains considerable oxygen. It is much inferior to hydro-carbon gas for fuel purposes, but is sometimes enriched by the addition of hydrogen from decomposing steam, when it is called water gas. The following table shows the average composition and available heat of combustion of the various fuels mentioned:

TABLE II.

FUEL	Per Cent.				t. u.
	Carbon	Hydr'gen	Oxygen	Ash	Ht. Val.
Dry wood	50	6	40	2	7800
Dry peat	58	6	30	5	10000
Lignite	69	5	20	6	13000
Bituminous	78	6	7	8	14000
Semi-bitum.	82	6	6	3	15000
Anthracite	92	2	3	3	13000
Petroleum	85	12	2	—	21000

A cord of dry wood is said to be equivalent to a ton of coal. Calling the density 40 lbs. per cubic foot and assuming the heat value of the coal as 14000 t. u. we would have:

$$\text{Heat value of wood} = \frac{2000 \times 14000}{128 \times 40} = 5500 \text{ t. u.}$$

16. Gaseous Fuels:—In the order of their value the gases used for fuel purposes are: natural gas, coal or illuminating gas, water gas, producer gas.

The following table given by Prof. Hutton shows the composition by volumes:

TABLE III.

		Natural Gas	Coal Gas	Water Gas	Prod'er Gas
Hydrogen		2.18	46.00	45.00	6.00
Marsh gas	CH_4	92.60	40.00	2.00	3.00
Carbonic oxide	CO	0.50	6.00	45.00	23.50
Olefiant gas	C_2H_4	0.31	4.00	——	——
Carbonic acid	CO_2	0.26	0.50	4.00	1.50
Nitrogen		3.61	1.50	2.00	65.00
Oxygen		0.34	0.50	0.50	——
Water vapor		——	1.50	1.50	1.00
Sulphydric acid		0.20	——	——	——

The following table shows the relative heat values of the four kinds of gas before named, and a comparison of each with soft coal. The coal is assumed to cost $2.50 per ton, and to have a net heat value, when burned under a boiler, of 10,000 heat units per pound. The gas is assumed to have an efficiency under the boiler of 80 per cent.

Comparison of Gas and Coal.

VARIETY	Heat Units per 1000 cu. ft.	Net Heat available 80 per cent.	Equivalent in pounds of coal.	Corresponding price per 1000 cu. ft.
Natural gas	1 000 000	800 000	80	10 cents
Coal gas	700 000	560 000	56	7 "
Water gas	350 000	280 000	28	3½ "
Producer gas	125 000	100 000	10	1¼ "

In the above calculation no attention has been paid to the greater convenience in handling gas. Conservative estimates show that a fireman can handle six or seven times as many horsepower of boilers with gas as with coal. Add to this the difference in freight and cartage, and you have a strong argument in favor of the use of gas.

Blast furnace gas comes practically under the same head as producer gas, having about the same chemical composition. It must, however, be cleansed before being used in an engine, since it contains dust and fine ash.

A similar comparison can be made for oil fuels, allowing the same efficiency of 80 per cent. A gallon of crude petroleum weighs about 6.8 lbs. and a gallon of fuel oil about 7.3 lbs. The latter is the heavier because some of the more volatile constituents of the crude oil have been distilled off in producing it.

	Heat Units per lb.	Heat Units per gal.	Net Heat per gal.	Equivalent in lbs. coal	Cor'sp'ding price c. per gal.
Crude oil	21 000	142 800	114 240	11.4	1.42
Fuel oil	18 000	131 400	105 120	10.5	1.31

17. Process of Combustion:—Carbon may burn incompletely to carbon monoxide CO, or completely to carbon di-oxide CO_2. In the former case only 4400 t. u. are given off, while in perfect combustion 14,500 t. u. are produced. The presence of CO in the gases of combustion is therefore a certain indication of waste.

The combustion may be complete at first and the CO_2 in passing over unburned coal may dissolve carbon with a loss of heat and form a certain amount of CO. This latter may be burned if additional air is supplied and the total heat will be the same as if all the carbon had been burned in the first place.

1 1-3 lbs. of oxygen are required per pound of carbon to produce CO and 2 2-3 lbs. to produce CO_2. As only 23 per cent of air by weight is oxygen, about 12 lbs. of air per pound of carbon is necessary to complete combustion. Ordinarily much more than this is supplied.

18. Hydrocarbons:—The heat of combustion of a hydrocarbon is nearly the same as the sum of the

heats of the contained hydrogen and carbon, if burned separately.

But if oxygen exists in the compound, enough of the hydrogen to form water with the oxygen will unite with the latter, and have no effect on the combustion.

Let the heat of combustion of one pound of hydrogen be 62032 t. u., and let C, H and O, be the per cent. by weight of each element in one pound of the combustible, then its total heat of combustion will be:

$$N = 145\ C + 620 \left\{ H - \frac{O}{8} \right\}$$

The other elements present in fuel are neutral and have practically no effect on the combustion. Carbon exists in most fuels in two forms, as fixed carbon and as allied with hydrogen in the volatile matter which distills from the fuel when it is heated.

Coals can be most readily classified by comparing the proportions of fixed carbon and volatile matter, thus:

COAL	Fixed Carbon	Volatile Matter	Moisture	Ash
Anthracite	83	4	3.5	8
Semi-Anthracite	83	9	1.0	6
Semi-Bituminous	75	20	1.0	3
Bituminous, Pa.	58	35	1.3	5
" Ohio	54	34	4.0	6
" Ill.	40	35	11.	13
Lignite, Iowa	35	37	8.	19
" Oregon	33	43	15.	8

The new and imperfectly formed coals, like the lignites, contain much moisture and volatile matter, while in the harder coals of the eastern states these have been driven off by heat and pressure.

19. Smoke:—The occasional production of dense black smoke is peculiar to that group of fuels known

as hydrocarbons, of which the more common are the petroleums and bituminous coal. The combustion of hydro-carbons seems to be always complete at first. If one watches the slow burning of a lump of cannel coal in the open grate he will see a whitish or yellowish vapor expelled from the coal by the gradual heat of the fire. This is the carbon and hydrogen combined which is distilled by the heat and leaves behind the free carbon as coke. While the escape of this vapor unburned represents a distinct loss of heat, the vapor is not smoke as we understand it. It does not deposit soot and will not stain or disfigure surfaces in its path.

As the heat increases and air is supplied the vapor ignites and burns with a yellow flame showing the presence of solid particles. If the temperature remains high and the air supply continues, the combustion is complete and the colorless carbon dioxide and water vapor pass up the chimney. If, however, the burning gas becomes chilled by contact with the relatively cool bricks of the chimney back or if insufficient air is supplied, the yellow flame becomes red and dingy, while particles of finely divided carbon are deposited on the adjacent surfaces or whirled away up the chimney.

20. Smoke Prevention:—The ordinary coal-oil lamp is one of the best illustrations of perfect combustion and consequent smoke prevention. The heated gases rising in the chimney produce a draft, and fresh air is continually drawn in at the bottom through the hot gauze, which warms and divides it so as to insure thorough mixing with the gases from the burning oil. Turn up the wick and the flame becomes smoky—too much hydro-carbon for the air supply. Raise the chimney slightly from the bottom and again there is smoke—too much air at too low a temperature, which chills the flame. Insert a cool metal rod into the chimney and soot is deposited on it—chilling of the flame again and disengagement of the carbon, while the hydrogen continues to burn.

And thus we may learn of the three requisites for good combustion; enough air, a sustained high temperature and a thorough mixing of the gases. The last two are so important that it is entirely possible to have an excessive supply of air and dense black smoke at the same time.

There are three methods of preventing smoke:

1.—The use of other fuels than soft coal:

The economy of a fuel depends on its cost and its evaporative power.

The first factor varies so widely for different localities that a general comparison of fuel economies cannot be made.

It is safe to say, however, that in most localities neither anthracite coal, coke or fuel gas can compete with soft coal. The only substitute which can be economically made for soft coal is crude petroleum, and that only on pipe lines.

2.—Varying supply of air:

If the amount of coal burned and of gas generated were always the same, it would be easy to supply just the right quantity of air at the right temperature to prevent smoke. In practice the rate of combustion is continually varying within a wide range, and must so vary with the ordinary hand-firing.

One class of smoke preventing devices vary the amount of air admitted over the grate, to correspond to the variation in the rate of combustion.

3.—Mechanical Stokers:

The object of the mechanical stoker is to feed the coal uniformly instead of intermittently and thus insure a uniform rate of combustion. Feeding coal in this way also prevents the periodical inrush of cold air over the grate, which is incident to the old method.

Many mechanical stokers are provided with steam jets above the fire which furnish additional oxygen.

An attempt to crowd the boiler beyond a certain

limit results in smoke, whatever method of firing may be used, and this is the chief cause of smoky chimneys in our cities. Moderate and careful stoking causes little smoke, but rushing the fire increases the nuisance.

19. Evaporative Power of Fuel:—To evaporate one pound of water at 212° F requires about 966 t. u. Dividing the total heat of combustion of a fuel by 966 gives therefore its theoretical evaporative power.

The average evaporative power of ordinary fuels in practice is about as follows:

TABLE IV.

NAME OF COAL	Evaporation per lb. of coal
Anthracite, broken . . .	10 lbs.
Anthracite, pea	9 lbs.
Cumberland, (bitum.) . .	11 lbs.
Ohio lump	9.5 lbs.
Gas coke	9 lbs.
Slack	6 to 8 lbs.
Crude oil	15 to 18 lbs.

20. Temperature of Fire:—To ensure complete combustion of a pound of carbon, 12 lbs. of air must be supplied if the mixture of the air with the burning gas is perfect.

In practice, 18 lbs. of air per pound of fuel are needed with forced draft, as fan or blast pipe, and 24 lbs. with ordinary chimney draft.

The resulting temperature of fire in each case may be found as in the following table taken from Rankine:

TABLE V.—Combustion of Carbon.

Air Used	Total Wt. of products	Mean Sp. Heat	Sp. Heat × Weight	Rise of Temperature
12 lbs.	13 lbs.	0.237	3.08	4708° F
18 lbs.	19 lbs.	0.237	4.51	3215° F
24 lbs.	25 lbs.	0.237	5.94	2440° F

The annexed table taken from "Steam" gives the temperature of a fire as determined by its appearance:

TABLE VI.—Temperature of Fire.

APPEARANCE	Temperature
Red, just visible	977°
Red, dull	1290°
Red, dull cherry	1470°
Red, full cherry	1650°
Red, clear cherry	1830°
Orange, deep	2010°
Orange, clear	2190°
White heat	2370°
White, bright	2550°
White, dazzling	2730°

21. Rate of Combustion :— The rapidity of combustion varies widely with circumstances. Ordinary rates are: For marine and stationary boilers, 10 lbs. to 20 lbs. per square foot of grate surface per hour; for locomotives, 75 lbs. to 150 lbs. per square foot of grate surface per hour.

EXAMPLES.

1.—Make a sketch and write a description of any mechanical stoker to which access can be had.

2.—Compare soft coal at $1.75 a ton, natural gas at 30 cents per thousand, illuminating gas at 75 cents per thousand and fuel oil at 2¼ cents per gallon.

3.—Determine the heat of combustion and the evaporative power of a coal containing by weight:

64.3% free carbon.
16.3% combined carbon.
5.8% hydrogen.
6.4% oxygen.
7.2% ash.

4.—Test the values given in the last column of Table V, page 16.

5.—A locomotive running at the rate of 45 miles an

hour has a grate surface of 15 sq. ft. Determine the maximum weight of coal and of water that would probably be required for a run of ninety miles.

6.—Determine the amount of heating surface and of grate surface needed under average conditions, in order that a boiler may evaporate 1500 lbs. of water per hour.

7.—Illustrate the phenomena of smoke production with a common kerosene lamp. Show why a chimney is used and why the gauze below the flame.

EXAMINATION.

1.—What is combustion and why is it accompanied by the production of heat?

2.—Name and describe principal natural fuels.

3.—What are some artificial fuels and how made?

4.—Classify varieties of coal and give heat values.

5.—Distinguish between different kinds of gas used as fuels.

6.—Distinguish between complete and incomplete combustion of carbon.

7.—What is meant by saying that the heat of combustion of a pound of carbon is 14,500 thermal units?

8.—Prove that about 70% of the heat is wasted with incomplete combustion.

9.—What is the effect of hydrogen and of oxygen in the compound?

10.—Is the term smoke consumer correct? Why?

11.—What is smoke and why is it formed?

12.—What is essential to prevention of smoke?

13.—What are the two principal functions of a mechanical stoker?

14.—What is the principal cause of smoke in manufacturing establishments?

15.—Explain how you would calculate the average temperature of a fire.

16.—What is meant by the evaporative power of a fuel and how is it calculated?

Chapter 3.

CHIMNEYS.

22. Chimneys :—Chimneys are required to carry off obnoxious gases and to produce draft. The size of the chimney depends upon the amount of gas to be carried off; the height upon the intensity of draft desired. Very high chimneys are expensive and unnecessary. About 150 feet is regarded as the economical limit. It is also better practice to build several small chimneys rather than one large one. When a large number of boilers or furnaces are dependent on one chimney those farthest from the chimney are likely to have poor draft.

The fuels having the longest flame require the least draft; wood less than coal and bituminous coal less than anthracite.

23. Pressure of Draft :—The pressure of draft is due to the difference between the weight of the column of hot gas inside the chimney and an equal column of cool air outside and may be expressed in pounds per square foot, in equivalent head of hot gas or in inches of water.

The volume of the furnace gases may, without sensible error, be taken the same as that of air at the same temperature. The volume of one pound of air at 32°F is about 12.4 cubic feet.

If the air supplied per pound of fuel be 12, 18 or 24 lbs., then will its volume at 32° be about 150, 225 or 300 cubic feet which we will call V_0.

The volume at any other temperature t is:

$$V = V_0 \frac{T}{T_0} = V_0 \frac{t+460}{492}$$

Let T_1 = absolute temperature of chimney gas.
T_2 = absolute temperature of outside air.
V_0 = volume of air per lb. of fuel or 300 cu. ft. at 32°F.

$$\frac{1}{V_0}=\text{lbs. fuel to one cu. ft. of air}=\frac{1}{300}=.0033.$$

$$\frac{1}{12.4}=.0807=\text{weight of one cu. ft. of air at } 32°\text{F}.$$

Then the weight of one cubic foot of chimney gas is:

$$\frac{492}{T_1}\left\{.0807+\frac{1}{V_0}\right\}$$

or usually $$\frac{492\times .084}{T_1}=\frac{41.33}{T_1}$$

The weight of a cubic foot of outside air is:

$$\frac{492\times .0807}{T_2}=\frac{39.7}{T_2}$$

Let H=height of chimney in feet.
p=pressure of draft in lbs. per sq. ft.
h=equivalent head of hot gas.
d=draft in inches of water.

Then the weight of column of gas in chimney one sq. ft. in area $= H\frac{41.33}{T_1}$ and weight of equal column of cool air $= H\frac{3.97}{T_2}$ Therefore $P=H\left\{\frac{39.7}{T_2}-\frac{41.33}{T_1}\right\}$

The height of a column of hot chimney gas which would produce the same pressure is obtained by dividing p by the weight of a cubic foot of this gas:

$$h=p\div\frac{41.33}{T_1}=H\left\{.961\frac{T_1}{T_2}-1\right\}$$

One inch of water $=\frac{62.5}{12}=5.21$ lbs. per square foot

$$d=\frac{p}{5.21}=H\left\{\frac{7.62}{T_2}-\frac{7.93}{T_1}\right\}$$

For a temperature of 600°F inside the stack, this formula gives about ¾ inch of water for each 100 feet of chimney.

24. Capacity of a Chimney: — The capacity of a chimney means the quantity of gas which it is capable

of delivering in a given time, and depends upon two factors: the area of the cross-section and the velocity of the gas. By the laws of fluids the velocity of the gas in feet per second will be $\sqrt{2gh}$ where h is the head of hot gas before mentioned.

Let A=area of flue in square feet.
v=velocity of gas in feet per second $=\sqrt{2gh}$.
w=weight of gas delivered per hour.
c=co-efficient of discharge.

Then w=3600 cAv×density of gas.

$$w=3600cA\sqrt{2gh}\times\frac{41.33}{T_1}=\frac{148800cA}{T_1}\sqrt{2gh}.$$

But $$h=H\left\{.961\frac{T_1}{T_2}-1\right\}$$

$$w=\frac{148800\ cA}{T_1}\sqrt{2gH\left\{.961\frac{T_1}{T_2}-1\right\}}$$

If we differentiate this regarding T_1 as a variable we have for a maximum of w:

$$T_1=\frac{2T_2}{.961}=2.081T_2$$

Substituting this value of T_1 in the above value for h;

$$h = H$$

and $$w=\frac{1194000\,cA\sqrt{H}}{T_1}$$

If we take 510 or 50°F as the value of T_2 then $T_1=$ 1061° or about 600°F, and $w=1125\ cA\sqrt{H}$ as the maximum rate of discharge.

So high a temperature in the stack is not however economical and a temperature of 400° would perhaps better represent modern practice.

Substituting this value in the formula there results:

$$w = 1090\ cA\sqrt{H}$$

The co-efficient c in this formula accounts for the losses due to friction and to cooling of the gas in the flue—Wm. Kent assumes that a layer of gas two inches thick next the surface of the flue is inert and has no

velocity; and derives the following formula: $E = A - 0.6\sqrt{A}$ as giving approximately the effective area of flue for either round or square chimneys.

The gas in a chimney is usually much cooler at the top than near the bottom, there being sometimes a difference of 100° in as many feet, and this reduces the discharge.

Allowing 24 lbs. of air to 1 lb. of fuel:

Let F=number pounds of fuel burned per hour $= \frac{W}{25}$

Then $F = 43.6\ cA\sqrt{H}$

Now experiment shows that the pounds of coal burned per hour under ordinary circumstances is more nearly $F = 15A\sqrt{H}$, or as given by Kent: $F = 16.65E\sqrt{H}$.

Assuming 5 lbs. of coal per boiler horse-power per hour we have: $HP = 3.33\ E\sqrt{H}$.

The loss of velocity by friction and from fall of temperature is seen to be very large, the coefficient of discharge being but little over one third. This may be easily tested by comparing the actual reading of a draft gage with the draft due to the temperature of the flue as shown by a pyrometer.

Mr. Wm. Wallace Christie gives the formula:

$$HP = 3.25\ A\sqrt{H}$$

and for area of grate:

$G = A\sqrt{H}$ for anthracite coal.

$G = 0.55\ A\sqrt{H}$ for bituminous coal.

The grate area is thus a function of the boiler horse power.

25. Height of Chimneys: — The following table is taken from Trowbridge's "Heat and Heat Engines," and is compiled from the results of experiments.

NOTE: The last four values in the table are taken from Thurston.

A comparison of the following values with those obtained from the formula: $F = 15A\sqrt{H}$, will show a tolerably close correspondence.

The character of the fuel, the type of boiler and

furnace used and the length of draft flues will all affect this problem and no general rule is possible.

TABLE VII.

Height of Chimneys and Rates of Combustion.

Height of Chimney in feet	lbs. of coal burned per sq. ft. of grate per hr.	Ratio of grate surface to chimney area is assumed as 8 : 1
20	7.5	
25	8.5	
30	9.5	
35	10.5	
40	11.6	
45	12.4	
50	13.1	
55	13.8	
60	14.5	
65	15.1	
70	15.8	
75	16.4	
80	16.9	
85	17.4	
90	18.0	
95	18.5	
100	19.0	
105	19.5	
110	20.0	
125	21.5	
150	23.5	
175	25.5	
200	27.5	

The highest chimney in the world is in Hutte, Saxony and is 460 feet high. The largest chimney is at the power house of the Metropolitan Street Railway in New York, being 353 feet high and 22 feet inside diameter.

EXAMPLES.

1. A chimney is 120 feet high and the temperature shown by the pyrometer is 428°F. Determine the draft in inches of water when the temperature of the external air is 40°F and when 90°F.

2. A chimney is 150 feet high, the temperature inside is 520°F and outside 60°F. Find the pressure of draft in ounces per square inch, in head of hot gas, in inches of water and in inches of mercury.

3. Determine the weight of gas discharged by the chimney in example 2, if it has a circular flue 48 inches in diameter.

4. Determine the nominal horse-power, consumption of coal and grate area for a chimney 150 feet high, having a circular flue 54 inches in diameter, if bituminous coal is burned.

5. Design a chimney for a plant where 1600 lbs. of bituminous coal are burned per hour, the area of grates being 80 square feet.

6. Determine the difference in effective area between a round and a square flue of the same area A, allowing dead layer of gas two inches thick.

Chapter 4.

THERMODYNAMICS.

26. **Thermodynamics** is the science which treats of the laws governing the relations between heat and mechanical energy.

The first law is thus enunciated by Rankine: "Heat and mechanical energy are mutually convertible; heat requires for its production and produces by its disappearance mechanical energy in the proportion of 778 ft. lbs. for each thermal unit."

This law is purely experimental. The ratio 778 is known as "*Joule's Equivalent*," from the name of its discoverer, and will be denoted by the symbol J.

Any quantity of heat may be expressed in foot-pounds by multiplying the number of thermal units by J.

Heat is usually converted into work by the expansion of some gas or vapor which loses heat energy as it performs mechanical work in expanding.

The area of an ordinary indicator diagram represents the amount of heat converted into work in one stroke of an engine.

Let h_1=no. of heat units received from boiler.

h_2=no. of heat units delivered to condenser.

Then $J(h_1-h_2)$=work done in foot pounds.

27. **Isothermal Expansion:** — When a gas expands at a constant temperature, it is said to expand isothermally, and the curve which represents graphically the corresponding pressures and volumes is called an isothermal curve, the different values of the pressure being taken as ordinates and the corresponding values of the volume as abscissae. It has been proved by experiment that if a perfect gas expand in this way, the product of the pressure by the volume remains constant or $pv=c$.

This is the equation of an equilateral hyperbola referred to its asymptotes as axes, and determines the character of the isothermal curve.

For common air $c=26214$ ft. lbs. at a temperature of 32°F.

28. Variable Temperature : — If the temperature of a perfect gas vary during expansion, then as has been shown in Article 7, the product of pressure and volume will vary as the absolute temperature, or : $pv=kT$.

Let p_0, v_0 and T_0 be respectively the pressure, volume and absolute temperature of one pound of dry air at 32°F.

$$p_0v_0=kT_0$$

and
$$k=\frac{p_0v_0}{T_0} \qquad \text{where :}$$

$$p_0=2116.3 \text{ lbs. per sq. ft.}$$
$$v_0=12.387 \text{ cu. ft.}$$
$$T_0=492°.$$
$$pv=\frac{p_0v_0}{T_0}T=53.27T$$

the equation for dry air.

29. Internal and External Work : —If a gas is heated at a constant volume, p will increase as T but no external work will be done and the heat used will all be expended in raising the temperature or in internal work. The specific heat of air at a constant volume $=C_v=0.169$ t.u,$=131.5$ ft. lbs.

Let the temperature of one pound of air be raised from T_1 to T_2 degrees absolute temperature ; then the heat expended will be : $Jh=131.5\,(T_2-T_1)$ ft. lbs.

On the other hand if a gas is expanded against a constant pressure, as that of the atmosphere, v will increase as T and external work of expansion will be done requiring additional heat.

Let the volume increase from v_1 to v_2 ; then will the external work be : $W=p(v_2-v_1)$.

for air $\quad pv=53.3T.$

and $\quad W=53.3(T_2-T_1).$

The total heat expended will be the external work + the internal work :

$$JH = W + Jh = 184.8\ (T_2 - T_1)$$

The constant 184.8 is sometimes called the specific heat of air at constant pressure, and will be denoted by C_p.

30. Adiabatic Expansion : When a gas expands without receiving or giving up heat, as in a non-conducting cylinder, it is said to expand adiabatically and the curve of pressures and volumes is called the adiabatic curve.

In this case, as no heat is received from outside, the work must be done at the expense of the internal energy or heat in the gas, and the temperature will fall.

Therefore the pressure will fall more rapidly than in isothermal expansion and the adiabatic curve may be represented by the equation : $pv^n =$ a constant, where n is greater than unity. See figure 3.

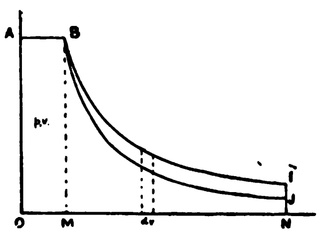


Fig. 3

31. Isothermal Curve :—The area under an isothermal curve represents graphically the work done by expansion of a gas at constant temperature.

Let this area BINM be called A, and let p=the pressure at any instant, and dv = infinitesimal change of volume. Then by calculus :

$$A = \int_{v_1}^{v_2} p\,dv \qquad \text{But} \quad pv = p_1 v_1 \text{ and } p = \frac{p_1 v_1}{v}$$

$$A = p_1 v_1 \int_{v_1}^{v_2} \frac{dv}{v} = p_1 v_1 (\log_e v_2 - \log_e v_1)$$

$$=p_1 v_1 \log_e \frac{v_2}{v_1} = kT_1 \log_e \frac{v_2}{v_1} \text{ for 1 lb. of gas.}$$

The whole area of the diagram

$$= p_1 v_1 + A = p_1 v_1 \left\{ 1 + \log_e \frac{v_2}{v_1} \right\}$$

This whole area represents the work done by admitting the volume of gas AB to a cylinder and then expanding it.

The mean pressure of the gas =

$$pm = \frac{\text{area}}{ON} = \frac{p_1 v_1}{v_2} \left\{ 1 + \log_e \frac{v_2}{v_1} \right\} = p_2 \left(1 + \log_e \frac{v_2}{v_1} \right)$$

32. **Adiabatic Curve:**—In this case

$$pv^n = p_1 v_1^n$$

and $p = p_1 v_1^n v^{-n}$

substituting this value of p in the equation

$$A = \int_{v_1}^{v_2} p\,dv$$

and we have

$$A = p_1 v_1^n \int_{v_1}^{v_2} v^{-n} dv$$

$$A = p_1 v_1^n \frac{v_2^{1-n} - v_1^{1-n}}{1-n}$$

$$A = p_1 v_1^n \frac{v_1^{1-n} - v_2^{1-n}}{n-1}$$

This expression may be reduced to either of three forms

$$A = \frac{p_1 v_1 - p_2 v_2}{n-1} = \frac{k}{n-1}(T_1 - T_2)$$

$$\text{or} \quad \frac{p_1 v_1}{n-1} \left\{ 1 - \left(\frac{v_1}{v_2} \right)^{n-1} \right\}$$

Adding the expression $p_1 v_1$ will give the whole area of the diagram $= \frac{np_1 v_1 - p_2 v_2}{n-1}$ The mean pressure equals this last divided by $v_2 = \frac{np_1 v_1 - p_2 v_2}{(n-1) v_2}$

33. **Work of Expansion:** – The total heat used in expanding a gas is in general:

$$JH = \text{internal work} + \text{external work.}$$

The internal work is always $= C_v(T_2 - T_1)$ where C_v is the specific heat of the gas expressed in foot pounds. In isothermal expansion $T_2 - T_1 = 0$, and therefore the internal work $= 0$.

Then the heat given to the gas is just equal to the external work done and the internal energy of the gas remains unchanged.

If the expansion is in a non-conducting cylinder, as no heat is received from outside, the external work is done at the expense of the internal energy.

$$JH = \text{internal work} + \text{external work}.$$

$$= C_v\left\{T_2 - T_1\right\} + \frac{p_1v_1 - p_2v_2}{n - 1}$$

$$= C_v\left\{T_2 - T_1\right\} + \frac{kT_1 - kT_2}{n - 1}$$

$$= \left\{C_v - \frac{k}{n-1}\right\}\left\{T_2 - T_1\right\}$$

per pound of gas.

But $\quad H = 0$ by assumption (adiabatic)

$\therefore \quad C_v - \dfrac{k}{n-1} = 0$

and $\quad n = \dfrac{C_v + k}{C_v} = \dfrac{C_p}{C_v}$

This is the ratio of specific heat at constant pressure to that at constant volume, and may be denoted by r.

Its value is as follows:

Dry Air....................	1.405
Saturated Air................	1.2
Superheated Steam...........	1.3
Saturated Steam.............	1,11
Ammonia Gas...............	1.27

The equation of the adiabatic curve may then be written

$$pv^r = p_1v_1^r$$

The density of saturated air is about double that of dry air and $k = 26.3$.

34. Fall of Temperature:—Let gas expand in a non-conducting cylinder from a pressure, volume and temperature denoted by p_1, v_1 and t_1 to those denoted by p_2, v_2 and t_2.

Then
$$p_2 v_2{}^r = p_1 v_1{}^r$$
$$p_2 v_2 = p_1 v_1 \left\{ \frac{v_1}{v_2} \right\}^{r-1}$$

But
$$pv = kT$$
$$kT_2 = kT_1 \left\{ \frac{v_1}{v_2} \right\}^{r-1}$$
$$T_2 = T_1 \left\{ \frac{v_1}{v_2} \right\}^{r-1}$$

or
$$\frac{v_1}{v_2} = \left\{ \frac{T_2}{T_1} \right\}^{\frac{1}{r-1}}$$

EXAMPLES.

1.—Prove in the equation for isothermal expansion, $pv = c$, that $c = 26214$ ft. lbs. for dry air at 32°.

2.—Find the value of pv at a temperature of 60° F; at 212° F.

3.—If 12 lbs. of dry air occupy a volume of 96 cu. ft. at a temperature of 90° F, determine the absolute pressure in pounds per square inch.

4.—Let 3 cu. ft. of dry air be expanded at atmospheric pressure by the addition of heat, the initial temperature being 60° F and the final 360° F. Determine: (1) The weight of air and its final volume. (2) The internal and external work in foot pounds.

5.—Compressed air at a pressure of 135 lbs. absolute expands to five times its original volume, in an engine cylinder. Determine the terminal pressure and the mean pressure. (1) If the temperature is kept constant. (2) If the cylinder is non-conducting and $r = 1.405$.

6.—Determine the final temperature of the air in Ex. 5, if expanding adiabatically, the initial temperature being 60° F.

7.—Saturated air at atmospheric pressure and a temperature of 70° F is compressed from a volume of 20 cu. ft. to a volume of 4 cu. ft. Determine: (1) The work of compression in foot pounds and the final pressure and temperature if the cylinder is non-conducting. (2) The quantity of heat to be abstracted by a water jacket to maintain a constant temperature. Draw diagram.

35. **Heat Diagrams**:—The diagrams in Fig. 3 show the amount of mechanical work done in expansion but give no information as to the heat energy present. To show the heat gained or lost in expansion the so-called temperature-entropy diagram is more convenient. Before considering this it will be necessary to explain what is meant by entropy.

36. **Entropy** of a gas is the quotient obtained by dividing its available heat energy or capacity for work by its absolute temperature.

Entropy is usually reckoned from 32° F as a zero and is denoted by the Greek letter φ.

As the product of pressure by volume in the indicator diagram gives the mechanical work in foot pounds, so the product of temperature by entropy in the heat diagram gives the heat energy in thermal units. The nature of entropy is not well understood and it cannot be measured directly. As temperature may be said to represent the intensity or voltage of the heat so entropy may be said to represent its quantity or current.

37. **Second Law of Thermodynamics**:—This law is variously stated by different authorities, but Rankine's enunciation is the best for our purpose. "If the absolute temperature of any uniformly hot substance be divided into any number of equal parts the effects of those parts in causing work to be performed are equal."

This is equivalent to saying that the conversion of heat energy into work is proportional to the change in absolute temperature, and is in accordance with the experiment described in Article 29, p. 26.

The isothermal curve represents expansion at a constant temperature and on the heat diagram is a horizontal straight line. The adiabatic curve represents expansion without loss or gain of heat or at constant entropy and is a vertical straight line on the heat diagram, showing no flow of heat.

38. Carnot Cycle:—In Figs. 4 and 5 are shown the heat and work diagrams respectively of a gas expanded and compressed in what is termed a Carnot cycle. From 1 to 2 the gas expands at a constant temperature T_1 the entropy increases from φ_1 to φ_2 and heat is received equivalent to the rectangle 12NM, (Fig. 4). From 2 to 3 the gas expands at a constant entropy φ_2 without gain or loss of heat (adiabatic) and the temperature falls to T_2.

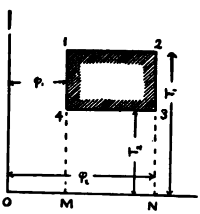


Figure 4

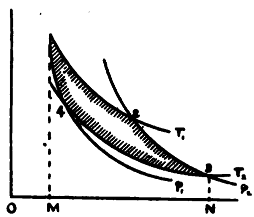


Figure 5

From 3 to 4 the gas is compressed isothermally to the original entropy and heat is lost as shown by rectangle 34MN (Fig. 4). From 4 to 1 the gas is compressed without change of entropy (adiabatic) to the original temperature T_1. The shaded area 1234 thus represents the excess of heat received or the energy converted into mechanical work.

$$H - h = (\varphi_2 - \varphi_1)(T_1 - T_2)$$

In Fig. 5 the curves T_1 T_2 are isothermal and the

curves φ_1 φ_2 are adiabatics. The ordinates are pressures and the abscissae are volumes so that the area 123NM represents positive work and the area 341 MN negative work. The shaded area 1234 consequently shows the net work done. As this area in Fig. 4 is expressed in heat units and in Fig. 5 is expressed in foot pounds, the numerical ratio of the second to the first will be 778 or $W = JH$ (conservation of energy).

39. **General Equations**:—Let a gas receive the minute quantity of heat JdH (expressed in foot pounds) and expand doing work $= pdv$.

Let $\quad dT =$ change of temperature.

$\quad d\varphi =$ change of entropy.

By definition $\quad Td\varphi = dH.$

From Art. 29 $\quad C_v dT =$ internal work.

$$JTd\varphi = JdH = C_v dT + pdv.$$

If expansion is adiabatic, $dH = 0$, therefore $d\varphi = 0$ and the entropy is constant as has been before stated.

The equation reduces to $pdv = -C_v dT$, or each increment of external work is done at the expense of an equal amount of internal energy. See Article 30.

This also shows that dT is negative and that the temperature is falling.

If the expansion is isothermal $dT = 0$ and the equation becomes $JTd\varphi = pdv$ and work is done by increase of entropy.

Dividing both members of the general equation by T:

$$Jd\varphi = C_v \frac{dT}{T} + \frac{pdv}{T}$$

But $\quad pv = kT \quad$ (Article 28)

and $\quad \frac{p}{T} = \frac{k}{v}$

Substituting $\quad Jd\varphi = C_v \frac{dT}{T} + \frac{kdv}{v}$

Integrating between limits

$$J(\varphi_2 - \varphi_1) = C_v \log_e \frac{T_2}{T_1} + k \log_e \frac{v_2}{v_1}$$

an equation which gives the change in entropy for any kind of expansion.

$k \log_e \frac{V_2}{V_1}$ may be also written $k \log_e \left(\frac{p_1}{p_2} \right) \frac{1}{n}$

Let $$\frac{V_2}{V_1} = R$$

Referring to Figs. 4 and 5 during the expansion 1 2, T is constant.

$$J(\varphi_2 - \varphi_1) = k \log_e R$$
$$JT_1(\varphi_2 - \varphi_1) = kT_1 \log_e R = \text{heat received.}$$

During the compression 3 4,

$$J(\varphi_2 - \varphi_1) = k \log_e \frac{V_3}{V_4} \quad . \quad . \quad . \quad . \quad \text{(Fig. 4)}$$

$$\therefore \qquad \frac{V_3}{V_4} = \frac{V_2}{V_1} = R$$

$$JT_2(\varphi_2 - \varphi_1) = kT_2 \log_e R = \text{heat lost.}$$
$$J(T_1 - T_2)(\varphi_2 - \varphi_1) = k \log_e R(T_1 - T_2) = \text{net heat received.}$$

40. **Constant Volume**:—As in Art. 29, let a gas be heated at a constant volume, no external work being done; we can show the heat expended, by means of the temperature-entropy diagram.

In the general equation

$$J(\varphi_2 - \varphi_1) = C_v \log_e \frac{T_2}{T_1} + k \log_e \frac{V_2}{V_1}$$

the last term disappears and we have an equation of φ in terms of T, from which a curve may be plotted as in Fig. 6.

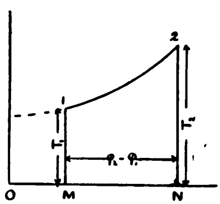


Fig. 6

The absolute values of φ depend on the zero of entropy which is assumed, and are of no particular consequence as it is the value $\varphi_2 - \varphi_1$ which determines the heat expended.

The line 12 is a logarithmic curve and the area 12NM shows the amount of heat used.

This area $= 131.5\ (T_2 - T_1)$ (see Art. 29). This curve must be used in finding the entropy of a gas above 32° F, since no external work is done.

41. Constant Pressure:—If the gas expands at a constant pressure when heated, the last term of the general equation may be transformed by inserting the relation

$$\frac{v_2}{v_1} = \frac{T_2}{T_1}$$

Then $J(\varphi_2 - \varphi_1) = C_v \log_e \frac{T_2}{T_1} + k \log_e \frac{T_2}{T_1} = C_p \log_e \frac{T_2}{T_1}$

The area 12NM in this case is $= 184.8(T_2 - T_1)$

This is the equation of a second logarithmic curve as shown in 12, Fig. 7, giving larger values of $\varphi_2 - \varphi_1$ and consequently larger values for the heat used for the same rise of temperature.

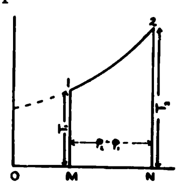


Figure 7

42. Isodynamic Expansion:—If a gas expand without doing any external work, as in the case of a gas expanding from one chamber through an opening into another chamber, then the intrinsic energy remains the same and the temperature is constant.

Heat may escape in two ways from a cylinder: as heat through the walls, or as mechanical energy through the piston-rod. This last is a case of no piston-rod.

This is sometimes called isodynamic expansion, but the curve is the same as the isothermal.

43. Total Heat of a Gas:—Let a gas of pressure and volume p_1 and v_1 expand indefinitely in a non-conducting cylinder, doing external work at the expense of its internal energy. Then will the indefinite area $1\varphi XM$ under the curve 1φ (Fig. 8) represent the external work done and therefore the heat expended.

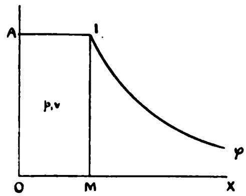


Figure 8

If the gas be supposed to expand to infinity, then will its temperature reach absolute zero,

$$T_2 = T_1 \left\{ \frac{v_1}{\infty} \right\}^{r-1} = 0$$

Consequently, all its internal heat will have been converted into work represented by the area under the expansion curve.

The total heat in a gas at a given pressure and volume may therefore be graphically represented by the area bounded by the initial ordinate, the axis of X and the adiabatic curve produced to infinity.

This area is $A = \frac{p_1 v_1}{r-1} \left\{ 1 - \left(\frac{v_1}{v_2} \right)^{r-1} \right\}_{v_2 = \infty} = \frac{p_1 v_1}{r-1}$

Total heat $= \frac{p_1 v_1}{r-1} = \frac{kT_1}{r-1}$ foot pounds $=$ JH.

$$\varphi_1 = \frac{H}{T_1} = \frac{k}{J(r-1)}$$

On the temperature-entropy diagram, the heat of the gas above 32° F is represented by the area included

between the ordinate, the axis of Y and the logarithmic curve of heating at constant volume. See Art. 40 and Fig. 6.

44. Change of Internal Heat:—The change of internal heat in a gas due to any change in pressure and volume, may be graphically represented by the area included between the curve of pressure and volume and two adiabatics drawn from its extremities and produced to infinity.

In Fig. 9, let 12 be any curve representing the change in pressures and volumes.

Draw the adiabatics $1\varphi_1$ and $2\varphi_2$ and suppose them to be infinitely extended.

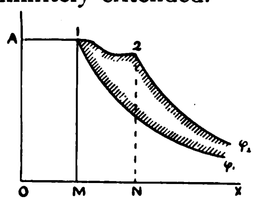


Figure 9

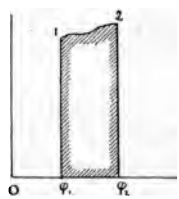


Figure 10

Let JH = initial energy of gas.
Jh = final energy of gas.
$\pm JH_1$ = heat received or rejected.
W = external work done.

Then $JH + JH_1 = W + Jh$ by law of conservation of energy, or $JH_1 = W + Jh - JH$.

But JH = area $1\varphi_1XM$.
Jh = area $2\varphi_2XN$.
W = area $12NM$.

$\therefore \quad JH = 12NM + 2\varphi_2XN - 1\varphi_1XM = \text{area } 12\varphi_2\varphi_1$.

This is much more readily shown by the temperature-entropy diagram as in Fig. 10 which is lettered to correspond to Fig. 9.

$$\text{The area } 12\varphi_2\varphi_1 = \int T d\varphi = \text{heat received.}$$

The difference between heat received and work done, *i.e.* between $12\varphi_2\varphi_1$, and $12NM$, is seen to be indepen-

dent of shape of the curve 12 and depends only on the location of the points 1 and 2.

45. Isothermal Expansion:—Let the curve of expansion in Fig. 9 be isothermal; then the total heat at 1 = area $1\varphi_1XM = \frac{p_1v_1}{r-1}$ as shown in Art. 43.

At 2, the total heat = area $2\varphi_2XN = \frac{p_2v_2}{r-1}$

But if expansion is isothermal, $p_2v_2 = p_1v_1$ and therefore the total heat or energy remains the same as has been before stated.

Consequently the heat received = the external work done, or in figure: $12\varphi_2\varphi_1 = 12NM$

To sum up: in isothermal expansion all work is done by heat received from outside, without impairing the intrinsic energy of the gas, while in adiabatic expansion, all work is done at the expense of the instrinsic energy.

46. Efficiency of Heat Engine:—Referring again to the so-called Carnot cycle as shown in Figs. 4 and 5 it has already been shown that the heat received during the expansion 1 2 is represented by the area 1 2 NM = $T_1(\varphi_2-\varphi_1)$ in Fig. 4. That the heat lost during the compression 34 is represented by the area 34 MN = $T_2(\varphi_2-\varphi_1)$ and finally that the shaded area 1234 = $(T_1-T_2)(\varphi_2-\varphi_1)$ represents the heat converted into work.

The ratio of this shaded area to the whole area 12NM is then the efficiency of the engine as a machine for converting heat into work.

This efficiency is evidently equal to $\frac{(T_1-T_2)(\varphi_2-\varphi_1)}{T_1(\varphi_2-\varphi_1)}$

or $$\text{Efficiency} = \frac{T_1-T_2}{T_1}$$

The efficiency of a simple heat engine is then the ratio of the fall in temperature of the gas to its initial absolute temperature and this is approximately true of any heat engine.

That no engine working between the same limits of temperature and entropy can be more efficient than the one referred to is evident from Fig. 11. Let the dotted line inscribed in the rectangle represent some other law of heat change.

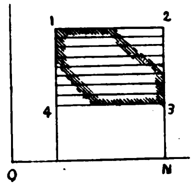


Figure 11

Then the included area which shows the energy converted into work is of necessity less than the area 1234. For a maximum efficiency it is then necessary that the heat should be taken in at one temperature and rejected at another temperature and that changes of temperature and entropy should not occur simultaneously. This condition is only partially realized in the steam engine.

47. Illustration of the Second Law:—Referring to the statement of this law in Art. 37 and to Fig. 11, it may be seen that the equidistant horizontal lines in the figure divide the area 1234 into equal parts.

Now if each interval represents a change of temperature of one degree, the corresponding rectangle represents the work done by reason of such change and all these amounts of work are equal, a verification of the second law.

Conversely, the work done by a simple heat engine following the Carnot cycle may be taken as a measure of the change in temperature and thus give us a thermometric scale which is independent of the coefficient of expansion or the specific heat of the substance. This relation is expressed by the formula already used: $H - h = (\varphi_2 - \varphi_1)(T_1 - T_2)$

EXAMPLES.

1.—What fraction of the total heat contained in the air would be converted into work in part (2) of Ex. 5 Art. 34?

Note that this is not a question of efficiency because not a complete cycle.

2.—Let one cubic foot of dry air at an initial temperature of 70° F be expanded and compressed in a Carnot cycle and let $\frac{v_2}{v_1}=2 \quad \frac{v_3}{v_2}=3$

(See Fig. 5.)

Determine:

(1) The lowest temperature.

(2) The change of entropy 12.

(3) The heat received and rejected and the efficiency.

3.—Work out the results for Ex. 2 with saturated air.

4.—Draw on cross section paper a diagram of the cycle in Ex. 2.

5.—Draw to scale temperature-entropy diagrams for constant volume and constant pressure as in Figs. 6 and 7 assuming the gas to be dry air heated from 60° F to 300° F.

6.—Let 4 cubic feet of dry air expand at a constant temperature of 80° F to double the volume. Determine the change in entropy and the amount of heat received. What is the work of expansion in foot pounds?

EXAMINATION.

1.—Explain the first law of thermodynamics in your own language.

2.—Give several illustrations of the conversion of heat into work and of work into heat.

3.—What is the general law governing the pressure, volume and temperature of a perfect gas?

4.—What is meant by a perfect gas?

5.—Explain the difference between heating the gas at a constant volume and at a constant pressure and give equations for dry air.

6.—Explain clearly the difference between isothermal and adiabatic expansion, as regards internal energy.

7.—Obtain an expression for the work done by expansion of a gas,

(1) At a constant temperature.

(2) In a non-conducting cylinder.

8.—Prove that the exponent r for adiabatic expansion is equal to $\frac{C_p}{C_v}$

9.—Find the fall of temperature when a gas expands in a non-conducting cylinder from v_1 to v_2.

10.—Explain what is meant by entropy and illustrate.

11.—Is entropy more mysterious than temperature? Why?

12.—Explain the second law of thermodynamics in your own language.

13.—Sketch a temperature-entropy diagram for a Carnot cycle and explain the relation of such a diagram to a work diagram.

14.—Deduce from the general equation $Td\varphi = dH$ an equation for a definite change of entropy $(\varphi_2 - \varphi_1)$ and draw a figure to illustrate.

15.—Show the forms which this equation takes for heating a gas at constant volume and constant pressure and sketch the temperature-entropy diagrams for each case.

16.—What can be shown by a temperature-entropy diagram that cannot be shown by an indicator card?

17.—How can the total heat in a gas be determined theoretically. Deduce an expression for this.

18.—Show the change in internal heat due to any

change of pressure and volume, by a work diagram and by a temperature-entropy diagram.

19.—On what does the efficiency of a heat engine depend? Prove this by a figure.

20.—Draw a temperature-entropy diagram which shall show the total heat in a gas above 32° F.

Chapter 5.

STEAM.

50. **Saturated Steam** is steam in contact with the water from which it was formed, and has always a temperature corresponding to its pressure. See table VIII.

Superheated steam is steam which has been heated above this temperature by being removed from contact with the water; each pound of superheated steam contains a fraction of a thermal unit for each degree elevation of temperature above that of saturated steam at the same pressure; it approaches the character of a perfect gas the more it is superheated.

Saturated steam is in a critical state and the slightest change in either its temperature or pressure causes either evaporation or condensation.

51. **Expenditure of Heat**:—In converting water into steam heat is expended in three ways: (1) Heating the water to the temperature of evaporation, commonly called the heat of the liquid, internal work. (2) Changing the water into steam, internal work of separating the molecules. (3) Expanding the volume from that of water to that of steam, external work against whatever may be the pressure.

The last two together are commonly termed the latent heat of evaporation or simply the latent heat. The sum of the three is called the total heat of the steam at that pressure and is commonly reckoned from 32° F. This is shown graphically in Fig. 1.

Let q = heat of the liquid per pound.
L = latent heat per pound.
H = total heat per pound.
t = temperature of evaporation in degrees F.

Then from experiments of Regnault :

$$H = 1091.7 + 0.305(t - 32).$$

$$H - L = q = t - 32 \text{ nearly.}$$

$$L = H - (t - 32) = 1091.7 - .695(t - 32).$$

or approximately :

$$L = 1092 - .7(t - 32).$$

$$= 966 - .7(t - 212)$$

Exercise:

Compare results given by these formulae with those in Table VIII.

52. Steam Curves:—If saturated steam expand isothermally, while in contact with the water, the pressure remains constant, giving a horizontal line on the work diagram, as when steam is admitted to a cylinder from a boiler.

After cut-off or separation of the steam from the water, if the steam be kept at a constant temperature by heat from a jacket, it will expand nearly as a perfect gas, and as the pressure falls and temperature remains constant, the steam will become superheated.

Saturated Steam Curve: If just enough heat is given to the expanding steam to keep it from condensing, its pressure and temperature will fall together as given in the tables, and the equation of the curve will be nearly:

$$pv^{\frac{17}{16}} = \text{a constant.}$$

a curve slightly below the isothermal.

Adiabatic for Steam: If the steam expand in a non-conducting cylinder, all the work being done at the expense of the internal heat, the temperature will fall still more rapidly, becoming less than that due to the pressure, and a part of the steam will condense to restore equilibrium.

A part of the work is then due to the latent heat of the condensed steam.

The resulting curve is very nearly:

$$pv^{\frac{10}{9}} = \text{a constant.}$$

being still lower on the diagram, as expansion goes on, than either of those before mentioned.

53. **Entropy of Steam**:—As the heat in saturated steam consists of two parts, the heat q, acquired during the rise of temperature of the water, and the heat L acquired at a constant temperature during evaporation, so may the entropy be regarded as consisting of two parts, the entropy of the water added during the first stage and the entropy of the steam added during the second stage.

The first is commonly denoted by θ and the second by φ.

The latter is easily calculated from the steam tables by dividing the heat of evaporation by the absolute temperature, or in symbols $\varphi = \frac{L}{T}$

54. **Entropy of Water**:—As the temperature is varying all the while that heat is added to the water it will be necessary to find the change of entropy by integration.

Let one pound of water be heated from the temperature, T_1 to the temperature T_2 and let the change of entropy be $\theta_2 - \theta_1$.

During any small change dT the change of entropy will be $d\theta$ and the heat added will be cdT, where c is the specific heat of water and varies slightly with the temperature.

$$d\theta = \frac{cdT}{T}$$

$$\theta = \int_{T_1}^{T_2} \frac{cdT}{T}$$

$$\theta_2 - \theta_1 = c \log_e \frac{T_2}{T_1}$$

(Compare with the formulas in Art. 39.)

The value of c is slightly greater than unity between 32° and 68° F, slightly less than unity between 68° and 105° F and then increases slightly to the boiling

point. It may, however, be taken as unity in ordinary calculations. Entropy is usually reckoned from 32° F and the values calculated from the formula calling $T_1 = 492$.

55. Indicator Diagrams:—The mechanical work done by the steam in an engine cylinder is clearly shown by the indicator diagram but this gives us no idea of the heat changes.

No mechanical apparatus has yet been invented which will automatically draw a temperature-entropy diagram and we are obliged to construct the latter from indicator diagram.

Let ABCD Fig. 12 represent the indicator diagram of an engine working under the following conditions:

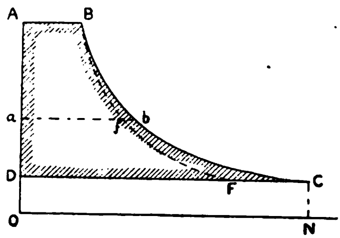


Figure 12

From D to A steam is admitted, the temperature and the pressure increasing; from A to B steam is admitted at a constant temperature T_1 and pressure p_1; from B to C the steam expands with temperature and pressure falling, receiving just enough heat to prevent condensation, *i. e.* BC is the curve for saturated steam corresponding to the values for p, t and d given in Table VIII; from C to D steam is expelled at a constant temperature T_2 and pressure p_2; the mechanical work done is now represented by the shaded area.

56. Heat Diagram for Steam:—The corresponding temperature-entropy diagram may now readily be constructed.

In Fig. 13 let abscissae represent value of entropy and ordinates represent absolute temperatures to any convenient scale. Then the point D will represent the temperature and entropy of the water at the lower limit $T_2\theta_2$. In a similar manner the point A shows

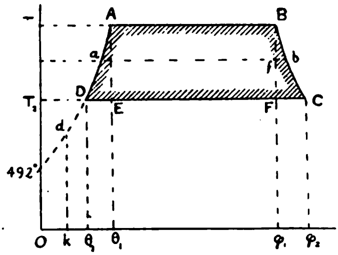


Figure 13

the condition of the water after heating to T_1 the rise of temperature being EA, the increase of entropy being DE and the heat added being shown bv the area

$$DA\theta_1\theta_2 = q_1 - q_2$$

The logarithmic curve DA may be plotted from the equation $\theta = c \log_e \frac{T}{492}$

In practice a straight line is a sufficiently close approximation.

Locate the point B so that AB may represent the entropy of steam at the higher temperature. Then the evaporation of the water at constant temperature is shown by the line AB, the increase of entropy being AB and the heat added being shown by the area AB $\varphi_1\theta_1 = L_1$ or the latent heat of evaporation at T_1.

Let DC be the entropy of dry steam at the lower temperature then will the curve BC represent the changes of temperature and entropy during the expansion in Fig. 12 and the area BC $\varphi_2\varphi_1$ will show the heat added to prevent condensation. BC also approximates closely to a straight line. The line CD represents condensation of the steam at the lower temperature which is equivalent to its expulsion dur-

ing exhaust and the area $CD\theta_2\varphi_2$ the amount of heat rejected. The shaded area accordingly represents the excess of heat received over that rejected and therefore the amount converted into work.

The efficiency is shown by the ratio

$$\frac{\text{area DABCD}}{\text{area DABC}\varphi_2\theta_2}$$

It is to be noticed that the heat shown by area $DAB\varphi_1\theta_2$ is furnished in the boiler and the heat $BC\varphi_2\varphi_1$ by the steam jacket. If the temperature of feed water is lower than that of the exhaust as shown by point d in Fig. 12, the area $dD\theta_2k$ must be added for this.

57. **Condensation**:—If no heat is supplied to the steam during expansion the expansion will be adiabatic as shown by the dotted line BF in either figure and the entropy will be constant.

T_2D still represents the entropy of the water at T_2 and as the entropy of the steam present is only DF instead of DC (Fig. 13) it follows that only the fraction $\frac{DF}{DC}$ now is steam and that the fraction $\frac{FC}{DC}$ has condensed during the expansion.

The efficiency under these new conditions is

$$\frac{\text{area DABFD}}{\text{area DAB}\varphi'_1\theta_2}$$

As the indicator diagram shows only the volume of steam present, we shall assume $\frac{DF}{DC}$ in Fig. 12 the same as $\frac{DF}{DC}$ in Fig. 13.

This indicates a method of determining points on the adiabatic curve for steam BfF in Fig. 12.

In Fig. 13 draw any horizontal line ab corresponding to a temperature T, then will $\frac{af}{ab}$ show the percentage of dry steam at that temperature for adiabatic expansion. On the diagram in Fig. 12 draw a hori-

zontal line ab corresponding to the pressure of the steam at a temperature T and on this line locate the point f so that the ratio $\frac{af}{ab}$ may be the same as in Fig. 13; then will f be a point on the adiabatic curve.

The diagram DABF is called the Rankine cycle, and is used for purposes of comparison with actual steam diagrams.

58. Diagram Without Expansion:—If in Fig. 12 the steam instead of expanding along the line BC is allowed to escape from the cylinder at B the resulting fall of pressure would be shown by a vertical line through B. The corresponding line BH on the entropy diagram, Fig. 14, is called a constant volume

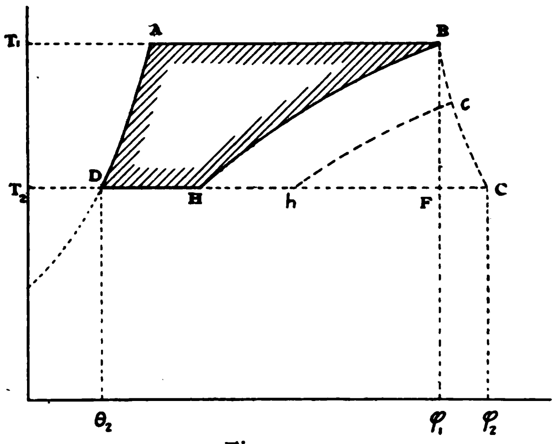


Figure 14

line and may be drawn in a manner similar to that already described for the line BF in Figs. 12 and 13. It may also be drawn directly by calculation from Table VIII. For instance, if the temperature at B is 350° F and that at C is 230° F the table shows the volumes of one pound of steam to be 3.32 and 19.01 cubic feet respectively.

Accordingly, if the volume remains constant at 3.32 cubic feet the final weight of steam will only be $\frac{3.32}{19.01}$ or 0.175 the initial weight. The entropy DH will accordingly be only 0.175 of DC the entropy of one pound at the lower temperature.

This is the case of steam working without expansion and the ratio $\frac{\text{work done}}{\text{heat received}} = \frac{\text{ABHD}}{\theta_2\text{DAB}\varphi_1}$ shows the low efficiency of such an arrangement. Similar constant volume lines as ch, Fig. 14, may be drawn from different points on the dry steam line BC to represent the effect of more or less complete expansion.

59. **Wet Steam**:—The steam in an engine cylinder is usually wet at cut-off on account of initial condensation. The steam from the boiler, at a temperature of over 300° F, comes in contact with the comparatively cool surfaces of the cylinder and piston heads which have just been exposed to the exhaust and considerable percentage is condensed and deposited as a film of water on those surfaces. This condensing process goes on throughout the stroke as fresh surfaces of cool metal are uncovered by the piston, but as the temperature and pressure of the expanding steam fall and the cylinder grows warmer, some of the water deposited at the first of the stroke is evaporated again.

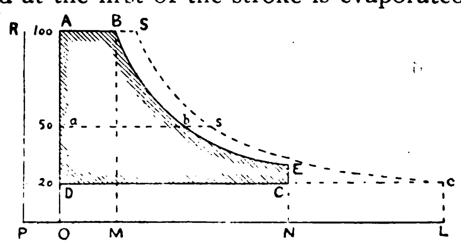


Figure 15

The actual amount of dry steam present at any point in the stroke can be determined by inspection of the indicator diagram. Usually with an early cut-

off more dry steam is found present at release than at cut-off.

In Fig. 15 let ABECD be the indicator diagram of an engine taking steam at 100 lbs. absolute and exhausting at 20 lbs. absolute. We will neglect the effects of wire drawing, compression and clearance.

Let AS be the volume which the steam would occupy at cut-off if dry, and let Ssc be the curve of saturation.

The quality of the steam at cut-off is $\frac{AB}{AS} = 75$ per cent. dry and at b the quality is $\frac{ab}{as}$ or about 80 per cent. dry; showing that the evaporation since cut-off has more than balanced the condensation.

In Fig. 16 draw the horizontal lines ABS and DCc corresponding to the absolute temperatures of admission and exhaust or 788° and 688° respectively.

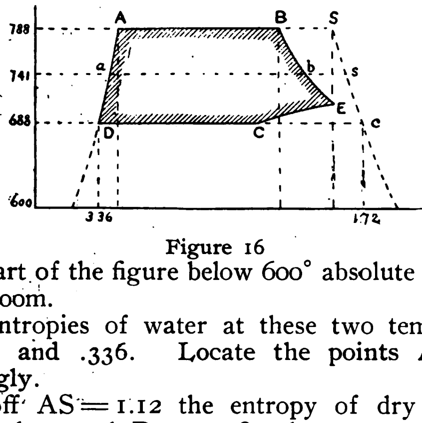


Figure 16

The part of the figure below 600° absolute is omitted to save room.

The entropies of water at these two temperatures are .473 and .336. Locate the points A and D accordingly.

Lay off AS = 1.12 the entropy of dry steam at 788° absolute and Dc = 1.384 the entropy of steam at 688°.

The temperature entropy diagram AScD corresponds to the complete expansion of dry steam as shown by the dotted lines in Fig. 15.

But, as explained in Art. 57 the percentage of entropy in the heat diagram corresponds to the percentage of dry steam present. Draw horizontal lines as abs Fig. 16 to represent temperatures corresponding to the various pressures in Fig. 15.

Locate the points B, b, E, C, etc., so that the ratios $\frac{AB}{AS}$, $\frac{DC}{Dc}$ etc., may equal those in Fig. 15 and plot the curves BE and EC.

Then will the shaded area ABECD represent the heat converted into work per pound of steam delivered by boiler.

The total heat received from the boiler will be represented by the area under DABS extending to the line of absolute zero plus heat used in heating feed water to 688°. The heat lost by condensation before cut-off is shown by the rectangle under BS, but a part of this is restored by the cylinder walls during expansion BE. The triangular area BbES shows the net heat lost by condensation in the cylinder. It will be understood that the point E may fall to the left or right of the vertical line through S. The efficiency is visibly less than if the steam had remained dry as per dotted lines.

60. Superheating:—The heating of steam after it has been removed from contact with the water raises its temperature to a point above that due to its pressure and the more it is thus superheated, the further is it removed from the critical point and the more it resembles a gas.

Such steam may accordingly be cooled down to the temperature of saturation without condensation, and this makes it a desirable medium for use in steam engine cylinders.

Fig. 17 shows the entropy diagram for steam which is superheated before expansion. As before, DA represents the heating of the water and AB the evaporation. The curved line BRS shows the change of

temperature and entropy due to superheating at a constant pressure, and the area under the line the amount of heat used in the process. Like DA the line BS is a logarithmic curve. (See Art. 54.)

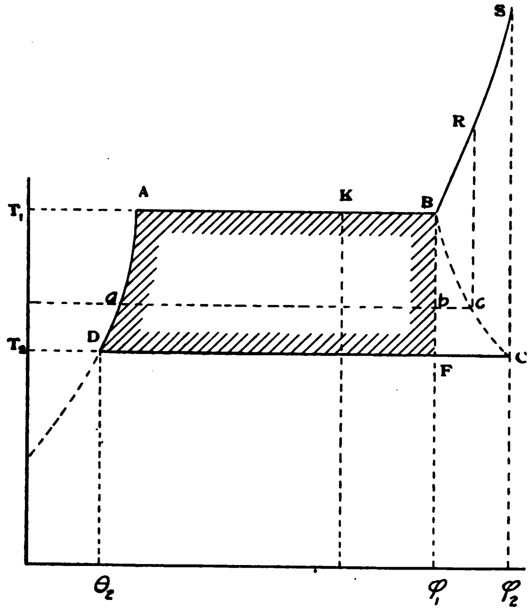


Figure 17

Until recently the value 0.48 has been accepted for the specific heat C_s of superheated steam.

Experiments conducted by the General Electric Co. and reported in "Power" for April, 1904, showed the following values for C_s:

Degrees of Superheat.	Specific Heat.
0	0.52
100	0.65
150	0.70
200	0.74
250	0.77

The steam pressure used was 155 pounds per square inch and the values given in the table are averages from different experiments.

A German experimenter reports values of C_s ranging from 0.468 at atmospheric pressure and no superheat up to 0.618 at 205 pounds pressure, and 190 degrees of superheat. In the absence of further data the values in the table will be used in this book.

If the steam expand adiabatically from S it will gradually lose its superheat until when it reaches the dry steam line at C it becomes again saturated steam.

If there is a less degree of superheat as at R and the steam expands as before, it becomes dry steam at C, and if the expansion continues some of the steam will condense.

Superheated steam on entering an engine cylinder usually cools at a constant pressure and the line BRS is retraced. If there is just enough heat to keep the steam dry the cooling stops at B and expansion begins.

Less superheat will allow the steam to condense along BK and expansion will begin at K, giving a diagram similar to Fig. 16.

The principal advantage of using steam is the prevention of this initial condensation in the engine cylinder.

61. Indicator Diagrams:—The actual steam curves in practice, as given mechanically by the steam engine indicator, vary widely in some cases from the theoretical curves on account of various disturbing causes such as:

1.—Friction of the steam in ports and passages.
2.—Condensation and radiation inside and outside the cylinder.
3.—Leakage of piston and valves.
4.—Clearance.
5.—Errors in the indicator itself and its attachment.

Friction: The effect of friction or wire-drawing of the steam as it is called, is to lower the pressure of

the steam, and is shown on the diagram by rounded corners, and by a fall of the admission line, and rise of the back pressure line. A part of this lost energy is restored as heat by friction.

The results of cylinder condensation have been shown in the preceding article. The effect on the indicator diagram is a fall of the steam line during the first part of the stroke on account of condensation, and a rise of the expansion line during the latter part of the stroke due to re-evaporation.

62. Leakage:—The leakage of steam by the admission valve causes a rise of the expansion line BE Fig. 15, and an apparent increase of efficiency, but a real loss. Leakage of steam by the piston has the effect of lowering the line ABE and raising CD. Any drop in the line AB may thus be due to wire-drawing, to initial condensation or to piston leakage.

An engine may readily be tested for leakage and the leaks located.

63. Clearance:—By clearance is meant the whole volume between the piston head when at the end of its stroke and the face of the admission valve.

Clearance is usually expressed as equivalent length of cylinder, or in per cent. of the piston displacement. It is shown on the diagram by moving the zero of volumes back from the end of the card, as to P, Fig. 15 and $\frac{OP}{ON}$ = the per cent. of clearance.

The actual volume of steam expanding is then PM and the actual ratio of expansion $= \frac{PN}{PM}$ which is less than $\frac{ON}{OM}$.

The pressure at the end of expansion will then be higher than when there is no clearance.

The amount of work done per unit of steam used, and therefore the efficiency, will diminish as the clearance increases. This effect may be neutralized somewhat by the judicious use of compression.

64. Indicator Errors:—The principal errors in the indicator itself are: friction or leakage of the indicator piston, defective or incorrect springs, inertia of the reciprocating parts of the indicator causing irregular lines in the diagram, faulty connection between the drum motion and the reciprocating parts of the engine, loss of pressure between the engine and the indicator due to unnecessarily long, narrow or tortuous passages. These defects may all be detected by examination and for the most part remedied.

65. Steam Consumption:—The actual amount of steam used may be determined by the condenser and weighing tank. The amount of dry steam present at different parts of the stroke may be determined from the indicator diagram.

Let the condition of the steam be assumed to be saturated as there is always water present, then the temperature and density at any point may be determined by use of the tables from the indicated pressure at that point. From this the weight of steam present may be calculated. The weights will all be determined per revolution.

Let $w =$ actual weight of steam by tank.
$x =$ indicated weight of steam at cut-off.
$y =$ indicated weight of steam at release.
$z =$ indicated weight of steam at compression.

If we assume steam to be dry at compression, $w + z$ $=$ mixture present in cylinder between cut-off and release.

As w represents the amount going through the cylinder and z the amount caught in the clearance, $w + z$ will be the total amount present after the steam valve closes and before the exhaust valve opens.

$$\frac{x}{w + z} = \text{per cent. of dry steam at cut-off.}$$

$$\frac{y}{w + z} = \text{per cent. of dry steam at release.}$$

$$y - x = \text{re-evaporation during expansion.}$$

Allowance should be made for initial moisture of the steam as shown by the calorimeter.

66. Theoretical Consumption:—The consumption of steam is usually expressed as so many pounds per I.H.P. per hour.

One I.H.P. for one hour $= 33000 \times 60 = 1980000$ ft. lbs.

The work which one pound of steam can do without expansion is:

Volume in cu. ft. × pressure in lbs. per sq. ft. $= P_1v_1$.

When the steam is used expansively the work becomes $W = P_1v_1(1 + \log_e R)$ where R is ratio of expansion.

The theoretical consumption of coal may then be expressed:

$$w = \frac{1\,980\,000}{P_1v_1(1+\log_e R)}$$

Example: Let $P_1 = 14400$ lbs. per sq. ft.

$R = 5$

From tables $v = \dfrac{1}{d} = 4.4$ cu. ft.

and $P_1v_1 = 63360$ ft. pounds.

$$w = \frac{1\,980\,000}{63\,360 \times 2.609} = 12 \text{ lbs.}$$

67. Consumption of Coal:—The consumption of coal in pounds per I.H.P. per hour may be determined as follows, on the assumption that all the heat of the coal could be converted into mechanical energy:

One pound of carbon in burning completely gives off 14500 t. u. or $14500 \times 778 = 11280000$ ft. lbs.

One I.H.P. for one hour $= 1980000$ ft. lbs.

$$\frac{1\,980\,000}{11\,280\,000} = .175 \text{ lbs. per I.H.P. per hour.}$$

68. The actual amounts of coal and water consumed, vary widely with different conditions, and the following table gives only good average results.

TABLE IX.

Consumption of Coal and Water per I. H. P. per hour.

KIND OF ENGINE	Coal	Water
High Duty Pumping . . .	1.2 to 2	12.5 to 18
Multiple Expansion Marine	1.4 to 2	14 to 20
Compound Condensing . .	1.6 to 2	17 to 20
Simple Condensing . . .	2 to 2.5	20
Simple Non-Condensing .	2.5 to 3.5	25 to 35
Locomotive	3 to 5	25 to 40

69. Flow of Steam:—The flow of steam from a boiler or cylinder at a higher pressure into a pipe or cylinder at a lower pressure is governed by the principle that, if radiation be neglected, the total energy of the steam will remain the same, since no external work is done.

Let p_1 and T_1 represent the pressure and temperature in the reservoir and q_1 and L_1 the corresponding heats.

Let p_2 T_2 q_2 L_2 represent the conditions in the pipe.

Suppose that the steam in the reservoir is x_1 per cent. dry and in the pipe x_2 per cent dry.

We will neglect the velocity of flow in the reservoir and call the velocity in the pipe u in feet per second. Then will the mechanical kinetic energy of a pound of steam in the pipe be $\frac{u^2}{2g}$

The general expression for the heat energy in wet steam is $JH = J(q + xL)$.

If no energy is lost by conduction and radiation we shall then have: $J(q_1 + x_1L_1) = J(q_2 + x_2L_2) + \frac{u^2}{2g}$.

(neglecting the volume of the condensed steam)

$$\frac{u^2}{2g} = J(q_1 - q_2 + x_1L_1 - x_2L_2).$$

Fig. 18 shows the changes of heat and entropy in the process, the shaded area corresponding to $\frac{u^2}{2g}$.

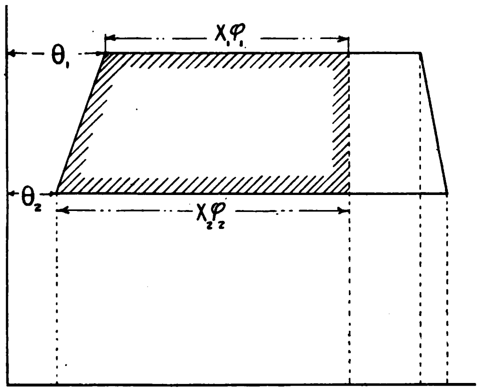


Figure 18.

To determine the relations between x_1 and x_2 we must remember that if no heat is lost or gained by conduction the change is adiabatic and therefore the entropy remains the same, or

$$x_1\varphi_1 + \theta_1 = x_2\varphi_2 + \theta_2$$

$$x_1\varphi_1 - x_2\varphi_2 = \theta_2 - \theta_1$$

But $\quad \varphi = \frac{L}{T} \quad$ and $\quad \theta_2 - \theta_1 = \log_e \frac{T_2}{T_1}$

$$\frac{x_1 L_1}{T_1} - \frac{x_2 L_2}{T_2} = \log_e \frac{T_2}{T_1}$$

$$x_2 L_2 = \frac{T_2}{T_1} x_1 L_1 - T_2 \log_e \frac{T_2}{T_1}$$

Substituting this value in the equation of energy and reducing, there results:

$$\frac{u^2}{2g} = J\left\{q_1 - q_2 + x_1 L_1 \left(1 - \frac{T_2}{T_1}\right) + T_2 \log_e \frac{T_2}{T_1}\right\}$$

From this we may determine u, the velocity of the steam.

As the steam is saturated all the while, the values of T and q can be determined from Table VIII when the values of p are known.

If the steam in the reservoir is nearly dry the sudden fall of pressure may cause the steam to become superheated.

Some experiments were made in 1886 at the Massachusetts Institute of Technology on the flow of steam through nozzles and orifices, and the results published. When the steam flowed through a brass tube 0.275 inches internal diameter and eight inches long, with entrance orifice rounded, the flow was 229 lbs. per hour with 69 lbs. gauge pressure at entrance and 4.4 lbs. at exit. Variation in the exit pressure up to 34 lbs. per square inch had little effect on the flow. The velocity calculated from this discharge would be 804 feet per second at the entrance.

Similar experiments with an orifice 0.25 inches in diameter in a thin plate, gave a discharge of 198 lbs. per hour with initial pressure of 72 lbs. and final pressure of 3.6 lbs. by gauge.

The calculated velocity would be 810 feet per second.

With an exit pressure of 51.6 lbs. the discharge was reduced to 141.5 lbs. per hour or a velocity of 576 feet per second.

The velocity of flow of superheated steam may be determined in a similar manner by taking the initial heat as: $J[H_1 + Cs(Ts - T_1)]$ where $(Ts - T_1)$ is the degree of superheat. Substituting this in the equation of energy, we have:

$$J[H_1 + Cs(Ts - T_1)] = J(q_2 + x_2L_2) + \frac{u^2}{2g}$$

Therefore $\dfrac{u^2}{2g} = J[H_1 + Cs(Ts - T_1) - q_2 - x_2L_2]$

70. **Weight of discharge:**—Napier's formula for the weight of steam flowing from an orifice is derived from the results of experiments and is closely correct when p_1 is equal to or greater than ten-sixths of the atmospheric pressure.

Let A be the area of the orifice in square inches. Then the discharge in pounds per second will be:

$$w = \frac{p_1 A}{70}$$

To find velocity from Napier's formula:

Let d = weight on one cubic foot of steam.
Q = volume in cubic feet per second.
v = velocity in feet per second.

Then $Q = \frac{Av}{144}$ and $w = Qd = \frac{Avd}{144}$

Therefore $\frac{Avd}{144} = \frac{p_1 A}{70}$ or $v = \frac{2p_1}{d}$ approximately.

71. **Flow in Long Pipes**:—Prof. Carpenter of Cornell University has recently made a series of experiments on the flow of steam in pipes, using diameters from ¾ inches up to 3 inches and various discharges up to 15000 lbs. per hour.

As a result of these experiments, he has proposed the following formula:

$$p = \frac{w^2 l \left(1 + \frac{3.6}{d}\right)}{7500\, D d^5}$$

in which p = fall of pressure in pounds per sq. in.
w = pounds of steam discharged per minute.
l = length of pipe in feet.
d = diameter of pipe in inches.
D = weight of a cubic foot of steam at lower pressure.

72. **Steam Injectors**:—Injectors are pumps used for forcing water into steam boilers, in which a part of the heat energy of steam is converted into mechanical energy for pumping the water, while the remainder is consumed in heating the water. The mechanical details of the injector will not be discussed here but simply the thermodynamic theory and the efficiency.

Let x equal the quality of the steam used, q and L its heats of liquid and of evaporation per pound.

Let one pound of this steam lift W pounds of feed water through a total height s, raise its temperature from t_1 to t_2 degrees Fahrenheit and deliver it to a boiler s′ ft. above the injector at a pressure of p pounds per square inch above the atmosphere, the velocity being v feet per second at the delivery nozzle.

The loss of heat in the steam expressed in foot pounds is $Jh = J(xL + q - q_2)$.

The heat given to the water is $Jh' = JW(q_2 - q_1.)$

The work of lifting the water will be Ws foot pounds and the work of forcing to the higher level of the boiler will be $(W + 1)s'$ foot pounds.

The kinetic energy of the water at the delivery nozzle will be $\frac{(W+1)v^2}{2g}$ and as this is the energy which is converted into the pressure work of forcing the water into the boiler, it must be equal to

$$\frac{144p(W+1)}{62.4} + (W+1)s' \text{ ft. pounds.}$$

$$\therefore \qquad \frac{v^2}{2g} = 2.31\,p + s'$$

i.e. velocity head = pressure head + lift.

If the thermodynamic efficiency is taken as unity, no allowance being made for loss by radiation, then

$$J(xL+q-q_2) = JW(q_2-q_1) + Ws + \frac{(W+1)v^2}{2g} \quad . \; . \; (a)$$

If we neglect the weight of the steam in the above equation we have for an approximate value:

$$W = \frac{J(xL + q - q_2)}{J(q_2 - q_2) + s + \frac{v^2}{2g}} \quad . \; . \; . \; . \; . \; . \; (b)$$

73. **Velocity of discharge**:—To determine the value of v we will assume the velocity of the steam before striking the water to be v = 800 ft. per sec. (See Arts. 69 and 70.)

Then, as the momentum after impact is the same as before we will have (neglecting the momentum of suction water):

$$\frac{800}{g} = \frac{(W + 1)v}{g}$$

or
$$v = \frac{800}{W + 1} \quad \ldots \ldots \quad (c)$$

The mechanical efficiency of the injector is small, since nearly all of the heat of the steam is expended in raising the temperature of the water.

The last two terms in the denominator of equation (b) are therefore relatively small and may be neglected in getting an approximate value of W.

With this value of W an approximate value of v is obtained in equation (c). Substituting this value of v in either (a) or (b) will give a value of W which is accurate enough for ordinary purposes.

It has been shown on p. 62 that $\frac{v^2}{2g}$ must be at least equal to $2.31\,p + s'$. If it be greater than this, the water will enter the boiler with some velocity.

If
$$v < 8\sqrt{2.31\,p + s'}$$
the injector will not work under the assumed conditions.

EXAMPLES.

1.—Determine the entropy of steam and of the water above 32° F at the pressure of 16, 28 and 75 pounds.

2.—Draw to scale, indicator and temperature-entropy diagrams for an engine taking steam at 115 lbs. absolute and rejecting it at 5 lbs. absolute, the ratio of expansion being 5.

3.—Determine from the diagrams in Ex. 2 the theoretical efficiency of the engine, and the amount of heat furnished by the jacket.

4.—Draw to scale a temperature-entropy diagram from an actual indicator card and interpret.

5.—Determine from the diagram in Ex. 4 the net loss of heat by condensation, assuming moisture at cut-off as twenty-five per cent.

6.—Determine from an actual indicator card the amount of dry steam present at cut-off, release and compression, allowing for clearance.

7.—Calculate the velocity and dryness of steam in a pipe leading from a boiler, the initial pressure being 105 lbs. and the final 90 lbs. absolute, the steam in the boiler being 90 per cent dry.

8.—Calculate the velocity of flow from the boiler in the preceding example if $p_2 = 65$ lbs. absolute; if 15 lbs. absolute.

9.—Calculate the weight of steam discharged in ten minutes from an orifice one inch in diameter into the air, if the initial pressure is 80 lbs. gauge. Compare with results of experiment.

10.—An engine uses 1200 lbs. of steam per hour at a pressure of 90 pounds gauge. The pipe joining the engine with the boiler is 172 feet long and 2 inches internal diameter. Required the pressure to be carried at the boiler.

11.—Calculate the probable weight of water per pound of dry steam in an injector which lifts water 6 feet, forces it 4 feet and raises its temperature from 80° F to 200° F, the boiler pressure being 90 lbs. absolute. Determine the velocity of entering the boiler.

EXAMINATION.

1.—Distinguish between superheated steam, saturated steam and wet steam.

2.—Can superheated steam be wet? Give reasons.

3.—Describe in your own language the process of converting cold water into superheated steam and draw a diagram to illustrate your meaning.

4.—Why is the entropy of steam divided into two parts.

5.—Determine the change of entropy of water in erms of the change in temperature.

6.—Describe an ideal apparatus which should draw a temperature-entropy diagram as the indicator draws a work diagram.

7.—Sketch an ideal indicator diagram and explain from it the changes of temperature and pressure.

8.—Sketch a temperature-entropy diagram to correspond to the above and explain what is meant by each line.

9.—Show in (8) just what heat is received in the boiler and what in the cylinder and show the heating of feed-water on the diagram.

10.—If the expansion is adiabatic what is the effect on the steam and how will it be shown in the diagrams?

11.—Sketch diagrams for expansion of wet steam and show from them the loss of efficiency.

12.—Describe in your own language the phenomena of condensation in a steam engine cylinder.

13.—Show by a temperature-entropy diagram the heat abstracted and restored by the cylinder walls and the net loss.

14.—Explain what is meant by clearance and its effect on the expansion line.

15.—Determine the consumption of coal per I.H.P. per hour with perfect efficiency.

16.—State the principal losses which reduce the efficiency of a complete plant.

17.—Calculate the velocity of flow of wet steam from a reservoir on the assumption that no heat is radiated.

18.—Calculate the percentage of dry steam $= x_2$ in the preceding question, at the lower pressure p_2.

19.—If the steam is dry in the reservoir, what will be its condition in the pipe?

20.—What is the available energy in a steam injector and in what four ways is it consumed?

21.—Apply the theory of equality of momenta to the injector.

22.—Distinguish between the thermodynamic and the mechanical efficiency of an injector and give an expression for each.

Chapter 6.

AIR, GAS AND REFRIGERATION CYCLES.

74. Air Compressor:—When air is compressed, stored and finally expanded in doing work, the only heat cycle is that due to the losses by cooling. Referring to Fig. 19, let air at pressure and volume A be compressed adiabatically to pressure and volume B, the air will be heated as shown in Art. 34. If the air could be kept hot until it was used in the engine it would expand down the adiabatic BA to the original pressure, volume and temperature and no heat would be lost. The usual cycle, however, is this: compression with rise of temperature along AB; cooling at constant pressure along BC to normal temperature; expanding and doing work from C to D with falling temperature; warming at atmospheric pressure to original condition at A.

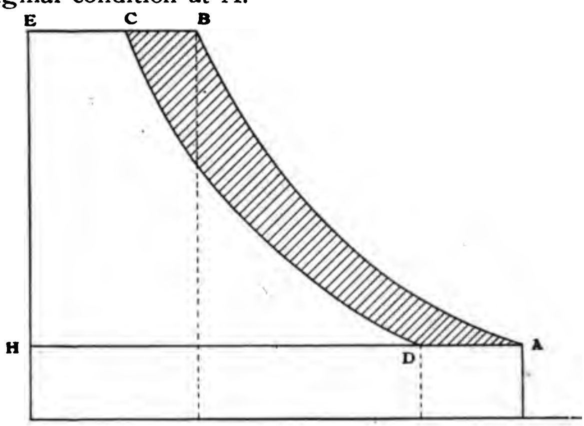


Figure 19

The area ABCD thus represents the loss due to radiation and conduction.

If it were possible to compress the air isothermally along AC, Fig. 20, rejecting a certain amount of heat and then utilize this same heat to give isothermal expansion, no heat would be lost, the compression and expansion being on the same line.

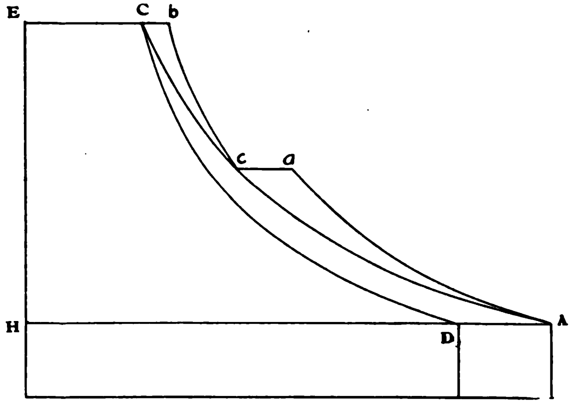


Figure 20

An approach to this is made by compressing the air in one cylinder to some point as a, cooling it to c, and finally compressing again to b in a second cylinder, Fig. 20.

If several cylinders were used in this way it is evident that the broken line would approach still closer to the isothermal AC and the heat loss be further reduced. In a similar manner two or three stages of expansion might be used, but this would involve the supply of heat from outside the system.

75. **Hot Air Engines**:—In hot air engines of the modern type the air is heated by a furnace and cooled by a water jacket or by coils. The action of the air can be understood from the diagram, Fig. 21, without considering the mechanism employed. Starting with air of low pressure and temperature at A, a compressor piston compresses the air to B isothermally since it is in contact with cold surfaces.

Leaving this cylinder the air passes through a hot regenerator to a larger working cylinder expanding by heat to the volume C. Cut-off takes place and the air expands isothermally in the large cylinder to D, being in contact with hot surfaces. DA shows the rejection of the air at a constant pressure.

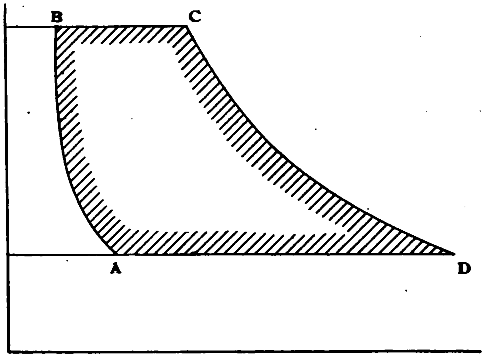


Figure 21

Fig. 22 is the corresponding entropy diagram with lettering arranged in the same order.

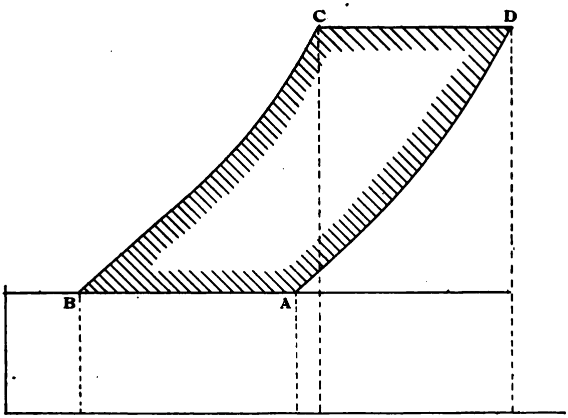


Figure 22

From A to B heat is rejected at a constant temperature; from B to C is the line for heating at a constant pressure (Art. 41, Fig. 7); from C to D heat is received at a constant temperature and from D to A is cooling at a constant pressure. The shaded area represents the net heat received and converted into work.

The efficiency of the hot-air engine is good, but it is very bulky in proportion to its power. Small engines of this class are much used for domestic pumping service.

76. Gas Engine Cycles:—The gas engine is simply a hot air engine in which the air is heated by the explosion of gas inside the cylinder of the engine itself.

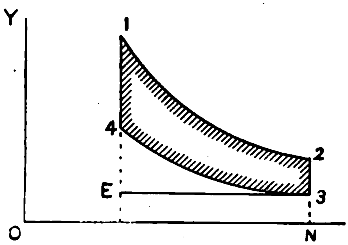


Figure 23

The diagrams are interesting from a thermodynamic standpoint and vary with the type of engine used. For illustration we will use the so-called Otto cycle, the diagram of the well-known Otto engine. Fig. 23 shows the cycle of operations which is as follows:

(1) The piston moves forward drawing in the explosive mixture through the admission valve at a constant pressure (line E 3).

(2) The piston returns compressing the mixture into the large clearance space OM, the admission valve being closed. (Line 3 4.)

(3) The mixture is ignited and explodes with a sudden increase of temperature and pressure. (Line 4 1.)

(4) The hot mixture expands along a curve 1 2 which is nearly adiabatic.

(5) The exhaust valve opens and the pressure suddenly falls (line 23) when the mixture is expelled from the cylinder at a constant pressure. (Line 3 E.)

Let

$$
\begin{aligned}
OM &= v_1 \\
ON &= v_2 \\
1M &= p_1 \\
2N &= p_2 \\
3N &= p_3 \\
4M &= p_4
\end{aligned}
$$

and let 12 and 34 be adiabatics for dry air.

Suppose for convenience that the engine uses one pound of the mixture per stroke. The work done is shown by the shaded area 1234.

$$\text{Area } 12NM = \frac{p_1v_1}{n-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{n-1}\right\}$$

See Art. 32.

$$\text{Area } 34MN = \frac{p_4v_1}{n-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{n-1}\right\}$$

$$\therefore \quad \text{Area } 1234 = (p_1-p_4)\frac{v_1}{n-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{n-1}\right\}$$

But $p_1 = p_4 \times \frac{T_1}{T_4}$ as the volume is constant during the heating, and $p_1 - p_4 = p_4\left\{\frac{T_1-T_4}{T_4}\right\}$ Substituting this value of $(p_1 - p_4)$ in the equation for work.

$$W = \frac{p_1v_1}{n-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{n-1}\right\}\frac{T_1-T_4}{T_4}$$

or the useful work done is proportional to the rise in temperature.

If we assume the specific heat of the mixture to be the same as that of air, the heat supplied in raising the temperature will be: $JH = 131.5(T_1 - T_4)$ ft. lbs. (See Art. 29.)

$$\text{The efficiency} = \frac{W}{JH} = \frac{\frac{p_4 v_1}{n-1}\left\{1-\left\{\frac{v_1}{v_2}\right\}^{n-1}\right\}}{131.5 T_4}$$

But by Art. 43 $\frac{p_4 v_1}{n-1} = 131.5\, T_4 =$ heat in the gas at T_4, *i.e.* $\frac{kT}{n-1} = C_v T$.

Therefore the efficiency of the cycle is :

$$E = 1 - \left\{\frac{v_1}{v_2}\right\}^{n-1} = 1 - \frac{T_2}{T_1}$$

As may be seen from the figure, the heat given to the gas at 41 is represented by the area between the curves 12 and 43 extended to infinity, consequently if the expansion were infinite or $\frac{v_1}{v_2} = 0$, all the heat would be converted into useful work.

The entropy diagram for the Otto cycle shows the difference between the efficiency of this and of the Carnot cycle. In Fig. 23a the lines 1 2 and 3 4 are

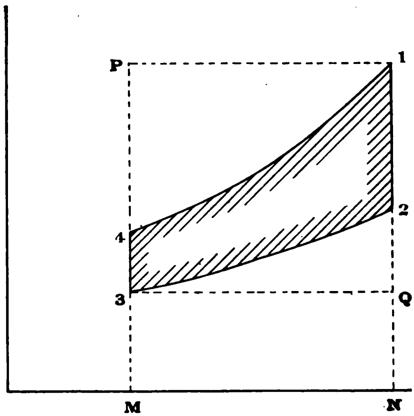


Figure 23 a

the constant entropy lines due to a non-conducting cylinder, while 2 3 and 4 1 show the cooling and

heating at constant volume (see Art. 40). The heat received is area $M41N = C_v(T_1 - T_4)$ and the heat rejected is area $N23M = C_v(T_2 - T_3)$. The heat converted into work is the difference shown by the shaded area and $= C_v(T_1 - T_2 + T_3 - T_4)$. The efficiency is therefore $= \dfrac{T_1 - T_2 + T_3 - T_4}{T_1 - T_4} = 1 - \dfrac{T_2 - T_3}{T_1 - T_4}$

But as the change of entropy is the same in both heating and cooling $\varphi_1 - \varphi_4 = C_v \log_e \dfrac{T_1}{T_4} = C_v \log_e \dfrac{T_2}{T_3}$

$$\text{and} \qquad \frac{T_1}{T_4} = \frac{T_2}{T_3} \qquad \therefore \qquad \frac{T_2}{T_1} = \frac{T_3}{T_4}$$

$$\therefore \quad \text{efficiency} = 1 - \frac{T_2}{T_1} \text{ as before.}$$

But the efficiency of the Carnot cycle shown in dotted lines would be : $1 - \dfrac{T_3}{T_1}$ or considerably more than the other.

The above is what is commonly called a four-cycle engine.

In the two-cycle engine there is an explosion at each revolution and the return stroke is employed in introducing the fresh charge and driving out the burnt gases at the same time.

The practical difficulty in accomplishing this has made two-cycle engines more or less unsatisfactory. The heat-process and the heat-diagram is the same as for the four-cycle engine.

77. **Actual Efficiencies**:—As might be expected the efficiency of the real gas engine is much less than that of the ideal on account of the loss of heat by radiation and conduction.

In practice it is found necessary to surround the cylinder with a water-jacket to carry off the heat which would otherwise be stored in the metal of the walls and raise the temperature to a dangerous degree.

Various experiments on Otto engines of from ten to fifteen horse-power show average working efficiencies about as follows:

Useful work.........	15 per cent.
Friction of engine....	5 per cent.
Loss at exhaust.......	25 per cent.
Heating water-jacket.	45 per cent.
Radiation	10 per cent.
	100

More heat may go out of the exhaust and less to the jacket but the amount converted into work is rarely over 20 per cent.

An analysis of a test on a ten-horse-power gasoline engine is reported in "Power" for May, 1903, and shows the following distribution of work in percentages:

Useful work....................	19.10
Friction of engine..............	3.70
Loss at exhaust.................	32.70
Heating water-jacket...........	38.50
Radiation	6.
	100.00

The proportion of gas to air by weight is as one to seven for maximum efficiency, according to experiments, and the consumption of gas may be from 20 to 25 cubic feet per indicated horse power per hour.

The value of n depends upon circumstances, but is usually from 1.3 to 1.4, the water jacket modifying the shape of the expansion and compression curves.

In actual diagrams the line 4 1 is inclined to the right, the explosion not being instantaneous, and the toe of the diagram is rounded by wire-drawing.

78. **The Diesel Engine**:—The Diesel oil engine is the result of an attempt to realize the high efficiency of the Carnot cycle. The success of this attempt may be judged by a comparison of diagrams.

The practical operation of the engine is as follows: Air is drawn in on the initial stroke and on the next stroke is compressed adiabatically to a high pressure, about 500 lbs. per square inch. This produces such a high temperature that when oil is pumped into the cylinder by an auxiliary pump, combustion at once begins. This combustion produces a further increase in temperature, but the oil is then pumped in just fast enough to maintain isothermal expansion until cut-off. After cut-off the expansion is adiabatic or nearly so, down to the terminal pressure.

The rejection of the burnt gases is the same as in the Otto engine.

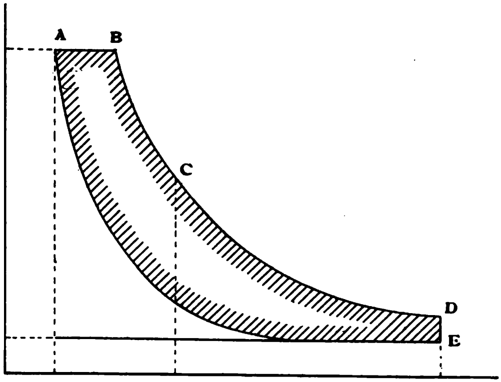


Figure 24

Figs. 24 and 25 show the pressure-volume and the temperature-entropy diagrams for this engine, similar letters showing corresponding points in the two.

The operations are as follows:

EA, compression of air accompanied by rise of temperature.

AB, injection of oil at constant pressure causing further rise of temperature.

BC, isothermal expansion due to further combustion.

CD, adiabatic expansion after cut-off.

DE, cooling at constant volume.

A comparison of Fig. 25 with Fig. 23 shows the advantage of the Diesel cycle to consist in the gradual combustion at a constant temperature shown by BC which makes the cycle approach more closely to the Carnot cycle EPCQ.

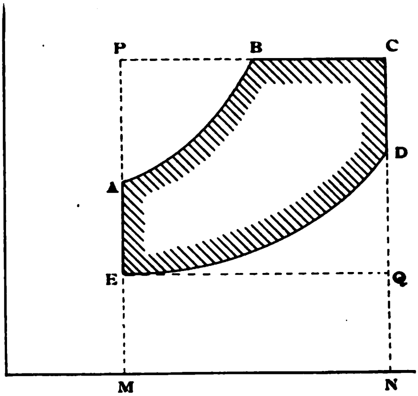


Figure 25

For the same temperature range the Diesel cycle has therefore a higher efficiency than the Otto.

Actual tests of this engine show efficiency to range from 23 per cent. at half load to 26.6 and 28.6 per cent. at full load. (Compare with Art. 77.)

Figs. 26 and 27 are from indicator diagrams of Otto and Diesel engines respectively. Various other gas

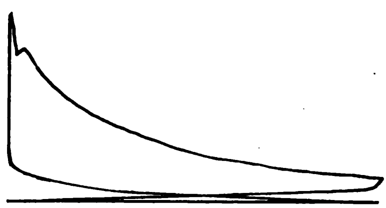


Figure 26

and air engine cycles, such as the Lenoir, the Joule and the Stirling are discussed in treatises on thermodynam-

ics, and are interesting from a scientific standpoint. As they do not represent the work of successful modern engines they are not introduced here.

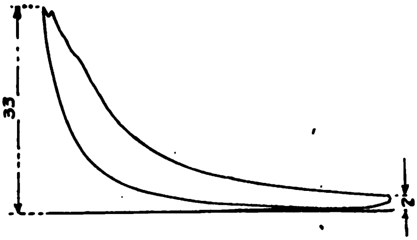


Figure 27

79. **Refrigerating Machinery**:—The object of a refrigerating plant is to abstract heat from some substance and thereby lower its temperature below that of surrounding bodies. Such a plant may be used to cool water, air or some other substance, but the principle in all cases is the same, and the process is the reverse of that used in the various heat engines which have been discussed in this book.

The object of a heat engine is to put heat into a gas or vapor and convert this into mechanical work. The object of a refrigerating machine is to draw heat from a substance by the exertion of mechanical energy. In the first case the object is to make the mechanical work as large as possible, in the second to keep it as small as possible.

In either case the gas or vapor is used merely as a medium of exchange, as a heat carrier, and the object of the process is to effect an interchange between heat and work. Air, being plentiful and cheap, has been tried as a medium in both processes, but neither air engines nor air refrigeration machines are convenient practically, on account of the large size of cylinders required and the great extremes of temperature which occur.

For a heat engine a medium is desired which has a boiling point above ordinary atmospheric temperatures and can be handled as a liquid. Water serves admirably for this purpose.

For refrigerating purposes a medium is best which has a boiling point below the normal temperature of the atmosphere and which under ordinary circumstances is a vapor. Only two such substances are in common use, ammonia and carbon dioxide. Probably 95 per cent. of refrigerating machines are of the ammonia type and only these will be considered here.

In order to understand the difference between the steam engine cycle and that of the ice machine, it will be well to study an ideal diagram of the two processes, as shown by Professor Hutton in his "Heat and Heat Engines."

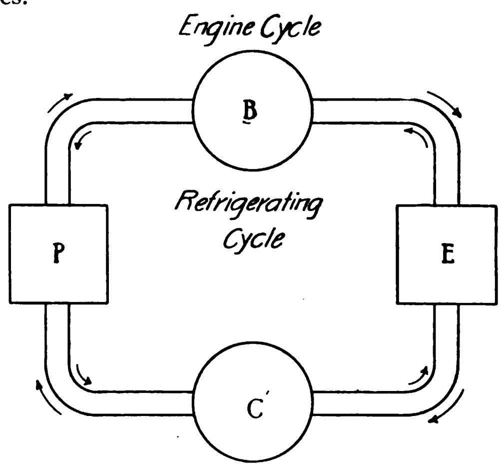


Figure 28

In Fig. 28 let the working fluid circulate through the pipes and reservoirs shown. Let B represent a receptacle which heats the fluid and C one which cools it. Let P represent a pump which carries the fluid from B to C or contra, and E a cylinder where expan-

sion takes place. If the machine is a heat-engine then the circulation will be clock-wise; B will be the boiler where heat changes the water to steam; E the engine where expansion of the steam converts heat into mechanical work, C the condenser where heat is abstracted from the steam and P the feed pump carrying water from the hot well to the boiler.

The object of the cycle is to do work at E and this is done by means of heat received at B.

If the machine is for refrigeration the cycle is reversed and becomes contra-clock-wise. The warm ammonia-vapor at a high-pressure goes from the compressor at P to the cooler C, where it is condensed to a liquid; E is the expansion valve where the liquid is allowed to vaporize and expand against the lower pressure beyond. B is the brine tank where the now cold vapor receives heat from the brine surrounding the coils, thereby cooling the latter, and finally P is the compressor which draws the vapor from B and compresses it into C.

The object of this cycle is to draw heat from B and the energy by which this is done comes from P.

From P around through C to E there is high pressure and from E around through B to P there is low pressure.

To pump from the low pressure to the high is the function of the compressor. Pure anhydrous ammonia is the fluid used as a heat-carrier. The characteristics of this substance are given in table X, which was originally calculated by Prof. DeVolson Wood.

As ammonia is a comparatively expensive material it is used over and over in the process and not rejected from the system, as is the steam from an engine. The line from E through B to P is the suction line, having a pressure varying from 5 to 20 lbs. gauge. This pressure depends upon the speed of the compressor and the opening of the expansion valve. For a temperature of 0° F in the cold storage at B, a pressure of

about 5 lbs. gauge is maintained, while a pressure of 20 lbs. corresponds to a temperature of about 32° F, there being usually 12 or 15 degrees difference between the temperature of the ammonia and that of the room cooled.

The ammonia vapor is compressed at P by a compressor mechanically driven and is delivered to C at a pressure of from 150 to 200 lbs. gauge and a temperature of from 70 to 100° F.

The vapor at this stage may be saturated or superheated. If liquid ammonia is present in the compressor the process is called wet and the temperatures will be limited to that corresponding to the pressure. On the other hand the vapor, if dry, will become superheated and the temperatures will range much higher, as in the so-called dry process. At C the vapor is cooled by running water and condensed. From this point it may be carried in uncovered pipes to any distance, as it is at ordinary atmospheric temperature. The expansion valve E is located near the brine tank or cold storage room, and here the liquid evaporates at the release of pressure, taking from the surrounding medium the latent heat of evaporation and expansion. At atmospheric pressure the latent heat of evaporation for ammonia is 573 t. u.

The standard unit of refrigeration as generally used is equivalent to the melting of 2,000 lbs. of ice at 32° F to water at the same temperature, or $142 \times 2000 = 284000$ t. u. On account of various losses the amount of energy actually used corresponds to about 420000 t. u. per ton of ice actually made.

The entropy diagram is the one most suitable for studying the refrigerating machine, since the process is primarily a thermodynamic one. As might be expected such a diagram, Fig. 29, is practically the same as that for a steam engine, but with a reversed cycle. Starting with the vapor at the lower temperature as it enters the compressor at A, we have compression which is more

or less adiabatic to B, superheating the vapor. If the vapor is wet when it reaches the compressors then compression may be along some line like a C with no superheat.

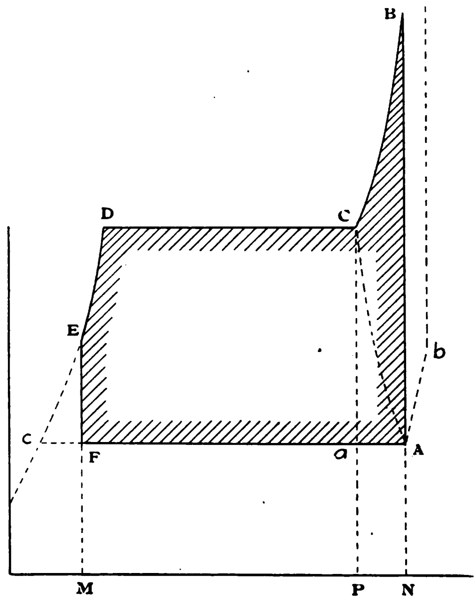


Figure 29

On the other hand the gas may become superheated and reach the compressor at some such condition as b. AC is the line of saturation for ammonia; distances to the right of this show superheat and distances to the left condensation. In a water-jacketed compressor the compression would not be adiabatic, but along some line approximating to AC and heat would be given up to the jacket water.

The specific heat of ammonia gas at constant pressure is 0.52.

BC is the line of cooling to the point of saturation, CD the condensation at the higher temperature, and DE the cooling of the liquid in the pipes.

At E the expansion valve is opened, EF is adiabatic expansion, accompanied by evaporation cF, and FA the evaporation at the lower temperature. If the expansion at E is free expansion with constant heat the line EF will incline slightly to the right and some heat will be used in reducing the liquid to boiling point. As has been shown the range of temperature will be ordinarily between 80° and — 15° F not including the peak at B.

The cycle is thus seen to be entirely below that of steam on the temperature scale.

The efficiency of the steam cycle is measured by the ratio of the shaded area to the entire area under the upper line. The object of improvements is to make the shaded area as large as possible, which is done by increasing T_1 and diminishing T_2.

In the refrigerating machine, on the contrary, the efficiency is measured by the ratio of the area under FA to the shaded area, since the former represents the heat extracted from the substance to be cooled and the latter the work expended. As Professor Reeve points out, this can hardly be called an efficiency, because these two areas are in a measure independent. The object, however, of improvements in this class of machinery would be to reduce the shaded area and the difference T_1—T_2 should be kept as small as practicable.

80. Actual Performance of Refrigerating Machines:—As has been already explained, the standard ton of refrigeration is 284000 t. u. and the actual making of one ton of ice usually takes from 410000 to 420000 t. u.

In a paper read before the American Society of Mechanical Engineers, Mr. J. C. Bertsch makes the

following recommendations for standard conditions as representing the averages obtaining in practice:

1. 284000 British thermal units as the latent heat of 2000 pounds of ice constitute one ton of refrigeration.

2. The efficiency of the ammonia compressor is 75 per cent. of the theoretical capacity.

3. The limit of piston travel in feet per minute shall be:

180 feet for strokes up to and including 12 inches.
240 feet for strokes over 12 and including 24 inches.
300 feet for strokes over 24 and including 36 inches.
360 feet for strokes over 36 inches.

4. The temperature to be produced is 15° F and the boiling point of the evaporating ammonia is zero.

5. The temperature of the condensing water is taken at 75° F and the temperature of the liquid ammonia at 80° F.

6. The displacement of the compressor must be 5 cubic feet or 8,640 cubic inches, per ton, per minute.

The calculation can then be made in this way:

Actual capacity of compression =

$$5 \times .75 = 3.75 \text{ cubic feet.}$$

$$= 3.75 \times .1094 = .41 \text{ lbs. at } 0°\text{F.}$$

$$555.5 - (80 \times 1.23) = 457 \text{ t. u. per pound.}$$

since specific heat of liquid $= 1.23$

$.41 \times 457 \times 24 \times 60 = 270\,000$ t. u. per day or practically one ton of refrigeration in twenty-four hours.

But in addition to the actual freezing of the water at 32° there is the cooling of the water from the temperature at which it enters the cans and the subsequent cooling of the ice to 15°. Assuming the water to be at a temperature of 80° F the heat per pound will be

$$(80 - 32) + \tfrac{1}{2}(32 - 15) = 56.5 \quad \text{t.u.}$$

$$2000 \times 56.5 = 113000 \text{ t. u. per ton.}$$

Adding this to the latent heat we have:

$$284000 + 113000 = 397000 \text{ t. u.}$$

External losses will bring the total up to an average of 415000 t. u. The actual ice-making capacity of the machine just calculated would then be about:

$$\frac{270\,000}{415\,000} = 0.65 \text{ ton in twenty-four hours.}$$

In the wet system of compression some liquid ammonia is present in the compressor and the line of compression may follow aC or some line to the left of this. Superheat and its attendant high temperature is thus avoided and the work of compression is reduced. Experience, however, shows very little difference in efficiency between this and the dry method.

The horse-power of the compressor under the conditions mentioned above would be approximately as follows:

Assume suction at 15 lbs. gauge and upper limit of pressure as 140 lbs. gauge, corresponding approximately to the limits of temperature already given.

Assume the compression as adiabatic and use the equation $pv^n = $ a constant.

n will be the ratio of $\frac{C_p}{C_v}$ and this for ammonia is

$$\frac{0.52}{0.41} = 1.27$$

$$\text{Ratio of compression} = \left\{\frac{155}{30}\right\}^{\frac{1}{n}} = 3.64$$

Initial volume = 5 cubic feet.

Final volume = 1.37 cubic feet.

Work of compression in foot pounds is by Art. 32:

$$A = \frac{np_2v_2 - p_1v_1}{n - 1}$$

$$= \frac{144(1.27 \times 155 \times 1.37) - 144(30 \times 5)}{.27}$$

$= 64000$ ft. lbs. per minute.

$=$ approximately two horse-power.

EXAMPLES.

1.—In a cycle like that shown in Fig. 19, find the work of compression and that of expansion in foot pounds, the heat lost in thermal units, and the efficiency of the operation.

Assume one pound of dry air to be used at 60° F and the initial and final pressures to be 15 lbs. and 105 lbs. absolute. State volumes and temperatures at A, B, C and D.

2.—Work out the same problem for the cycle shown in Fig. 20, assuming that the air is cooled to original temperature at 60 lbs. pressure.

3.—Draw to scale an entropy diagram for the cycle in Fig. 19.

4.—If the hot-air engine, whose cycle is shown in Fig. 21, works between the pressures of 15 ibs. and 105 lbs. and the temperatures 100° and 500° F, find the foot pounds of work developed per pound of dry air.

5.—Find the efficiency of the cycle from Fig. 22, using the same data as in example 4.

6.—A gas engine working on an Otto cycle like that in Fig. 23, shows the following gauge pressures on the indicator card: 1 = 270 lbs. 2 = 42 lbs. 3 = 0 lbs. 4 = 60 lbs.

If the expansion and compression are adiabatic and the gas enters at 60° F, find the temperatures at 1, 2 and 4. (See diagram, Fig. 26.)

7.—Find the efficiency of the cycle in preceding example and compare with efficiency of Carnot cycle under similar conditions.

8.—Give the equation of each of the lines in Fig. 25.

9.—In Fig. 29 find the heat abstracted from the brine tank, the heat carried away by the condensing water, and the net heat due to mechanical work. Assume the

ammonia to be dry at A and C and the temperatures to be as follows: at A, — 15° F; at C, 80° F; at E, 60° F. Assume one pound of ammonia used and work with and without superheat.

EXAMINATION.

1.—Draw cycle of air compressor and engine and show why work is lost.

2.—Explain why two-stage compression with an inter-cooler reduces the loss.

3.—Draw pv diagram for hot-air engine and explain changes.

4.—Draw entropy diagram for preceding and show different heat quantities.

5.—Sketch and describe the Otto cycle for a gas engine.

6.—Determine the formula for work and the efficiency in the Otto cycle.

7.—Prove the formula for efficiency by means of the entropy diagram.

8.—Show in what ways the heat is used in an actual engine and the approximate percentages.

9.—Show in what way the Diesel cycle differs from the preceding.

10.—Compare the entropy diagram with that of the Otto cycle and show which is more efficient.

11.—Describe briefly the function of a refrigerating machine and how it differs from that of a heat engine.

12.—Draw a diagram showing the members of the two kinds of machines and trace the two cycles.

13.—Trace in detail the different operations in an ammonia refrigerating machine, giving pressures, temperatures, etc.

14.—Draw entropy-diagram for this type of machine and explain all lines.

15.—Distinguish between wet and dry processes on the entropy diagram.

16.—What is meant by the efficiency in the refrigerating machine and how is it increased?

17.—Explain what is meant by unit of refrigeration and by ton of refrigeration.

18.—Give abstract of prevailing practice as to capacity, efficiency and range of temperature in ice machines.

19.—Show by calculation how the capacity of a one-ton machine may be figured.

20.—Prove that such a machine would only produce about two-thirds of a ton of ice in 24 hours.

TABLE VII.

Table of Hyperbolic Logarithms, ($\log_e$)

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.05	0.049	2.65	.975	4.25	1.447	5.85	1.766	7.45	2.008
1.10	.095	2.70	.993	4.30	1.459	5.90	1.775	7.50	2.015
1.15	.140	2.75	1.012	4.35	1.470	5.95	1.783	7.55	2.022
1.20	.182	2.80	1.030	4.40	1.482	6.00	1.792	7.60	2.028
1.25	.223	2.85	1.047	4.45	1.493	6.05	1.800	7.65	2.035
1.30	.262	2.90	1.065	4.50	1.504	6.10	1.808	7.70	2.041
1.35	.300	2.95	1.082	4.55	1.515	6.15	1.816	7.75	2.048
1.40	.336	3.00	1.099	4.60	1.526	6.20	1.824	7.80	2.054
1.45	.372	3.05	1.115	4.65	1.537	6.25	1.833	7.85	2.061
1.50	.405	3.10	1.131	4.70	1.548	6.30	1.841	7.90	2.067
1.55	.438	3.15	1.147	4.75	1.558	6.35	1.848	7.95	2.073
1.60	.470	3.20	1.163	4.80	1.569	6.40	1.856	8.00	2.079
1.65	.500	3.25	1.179	4.85	1.579	6.45	1.864	8.05	2.086
1.70	.531	3.30	1.194	4.90	1.589	6.50	1.872	8.10	2.092
1.75	.560	3.35	1.209	4.95	1.599	6.55	1.879	8.15	2.098
1.80	.588	3.40	1.224	5.00	1.609	6.60	1.887	8.20	2.104
1.85	.612	3.45	1.238	5.05	1.619	6.65	1.895	8.25	2.110
1.90	.642	3.50	1.253	5.10	1.629	6.70	1.902	8.30	2.116
1.95	.668	3.55	1.267	5.15	1.639	6.75	1.910	8.35	2.122
2.00	.693	3.60	1.281	5.20	1.649	6.80	1.917	8.40	2.128
2.05	.718	3.65	1.295	5.25	1.658	6.85	1.924	8.45	2.134
2.10	.742	3.70	1.308	5.30	1.668	6.90	1.931	8.50	2.140
2.15	.765	3.75	1.322	5.35	1.677	6.95	1.939	8.55	2.146
2.20	.788	3.80	1.335	5.40	1.686	7.00	1.946	8.60	2.152
2.25	.811	3.85	1.348	5.45	1.696	7.05	1.953	8.65	2.158
2.30	.833	3.90	1.361	5.50	1.705	7.10	1.960	8.70	2.163
2.35	.854	3.95	1.374	5.55	1.714	7.15	1.967	8.75	2.169
2.40	.875	4.00	1.386	5.60	1.723	7.20	1.974	8.80	2.175
2.45	.896	4.05	1.399	5.65	1.732	7.25	1.981	8.85	2.180
2.50	.916	4.10	1.411	5.70	1.740	7.30	1.988	8.90	2.186
2.55	.936	4.15	1.423	5.75	1.749	7.35	1.995	8.95	2.192
2.60	.956	4.20	1.435	5.80	1.758	7.40	2.001	9.00	2.198

TABLE VIII. Properties of Saturated Steam.

Absolute pressure lbs. per sq. inch	Temperature of boiling point degrees F	Heat of the liquid from 32° F	Total heat from 32° F	Latent heat	Weight of a cub. ft. in pounds	Cu. ft. per pound
p	t	q	H	L	d	$\frac{1}{d}$
1.0	102.0	70.0	1113.1	1043.0	0.00299	334.6
2.0	126.3	94.4	1120.5	1026.1	0.00576	173.6
3.0	141.6	109.8	1125.1	1015.3	0.00844	118.4
4.0	153.1	121.4	1128.6	1007.2	0.01107	90.31
5.0	162.3	130.7	1131.5	1000.8	0.01366	73.22
6.0	170.1	138.6	1133.8	995.2	0.01622	61.67
7.0	176.9	145.4	1135.9	990.5	0.01874	53.37
8.0	182.9	151.5	1137.7	986.2	0.02112	47.07
9.0	188.3	156.9	1139.4	982.5	0.02374	42.13
10.0	193.2	161.9	1140.9	979.0	0.02621	38.16
11.0	197.8	166.5	1142.3	975.8	0.02866	34.88
12.0	202.0	170.7	1143.6	972.9	0.03111	32.14
13.0	205.9	174.6	1144.7	970.1	0.03355	29.82
14.0	209.6	178.3	1145.8	967.5	0.03600	27.79
14.7	212.0	180.7	1146.6	965.8	0.03758	26.64
15.0	213.0	181.8	1146.9	965.1	0.03826	26.15
16.0	216.3	185.1	1147.9	962.8	0.04067	24.59
17.0	219.4	188.3	1148.9	960.6	0.04307	23.22
18.0	222.4	191.3	1149.8	958.5	0.04547	22.00
19.0	225.2	194.1	1150.7	956.6	0.04786	20.90
20.0	227.9	196.9	1151.5	954.6	0.05023	19.91
21.0	230.5	199.5	1152.3	952.8	0.05259	19.01
22.0	233.1	202.0	1153.0	951.0	0.05495	18.20
23.0	235.5	204.5	1153.7	949.2	0.05731	17.45
24.0	237.8	206.8	1154.4	947.6	0.05966	16.76
25.0	240.0	209.1	1155.1	946.0	0.06199	16.13
26.0	242.2	211.2	1155.8	944.6	0.06432	15.55
27.0	244.3	213.4	1156.5	943.1	0.06666	15.00
28.0	246.4	215.4	1157.1	941.7	0.06899	14.49
29.0	248.3	217.4	1157.7	940.3	0.07130	14.03
30.0	250.3	219.4	1158.3	938.9	0.07360	13.59
31.0	252.1	221.3	1158.8	937.5	0.07590	13.18
32.0	254.0	223.1	1159.4	936.3	0.07821	12.78
33.0	255.8	224.9	1159.9	935.0	0.08051	12.41
34.0	257.5	226.7	1160.4	933.7	0.08280	12.07
35.0	259.2	228.4	1161.0	932.6	0.08508	11.75

Properties of Saturated Steam (continued)

Absolute pressure lbs per sq inch	Temperature of boiling point degrees F	Heat of the liquid from 32° F	Total heat from 32° F	Latent heat	Weight of a cub. ft. in pounds	Cub. ft. per pound
p	t	q	H	L	d	$\frac{1}{d}$
40.0	267.1	236.4	1163.4	927.0	0.09644	10.37
45.0	274.3	243.6	1165.6	922.0	0.1077	9.287
50.0	280.8	250.2	1167.6	917.4	0.1188	8.414
55.0	286.9	256.3	1169.4	913.1	0.1299	7.696
60.0	292.5	261.9	1171.2	909.3	0.1409	7.096
65.0	297.8	267.2	1172.7	905.5	0.1519	6.583
70.0	302.7	272.2	1174.3	902.1	0.1628	6.144
75.0	307.4	276.9	1175.7	898.8	0.1736	5.762
80.0	311.8	281.4	1177.0	895.6	0.1843	5.425
85.0	316.0	285.8	1178.3	892.5	0.1951	5.125
90.0	320.0	290.0	1179.6	889.6	0.2058	4.858
95.0	323.9	294.0	1180.7	886.7	0.2165	4.619
100.0	327.6	297.9	1181.9	884.0	0.2271	4.403
105.0	331.1	301.6	1182.9	881.3	0.2378	4.206
110.0	334.6	305.2	1184.0	878.8	0.2484	4.026
115.0	337.9	308.7	1185.0	876.3	0.2589	3.862
120.0	341.0	312.0	1186.0	874.0	0.2695	3.711
125.0	344.1	315.2	1186.9	871.7	0.2800	3.572
130.0	347.1	318.4	1187.8	869.4	0.2904	3.444
135.0	350.0	321.4	1188.7	867.3	0.3009	3.323
140.0	352.8	324.4	1189.5	865.1	0.3113	3.212
145.0	355.6	327.2	1190.4	863.2	0.3218	3.107
150.0	358.3	330.0	1191.2	861.2	0.3321	3.011
155.0	360.9	332.7	1192.0	859.3	0.3426	2.919
160.0	363.4	335.4	1192.8	857.4	0.3530	2.833
165.0	365.9	338.0	1193.6	855.6	0.3635	2.751
170.0	368.3	340.5	1194.3	853.8	0.3737	2.676
175.0	370.6	343.0	1195.0	852.0	0.3841	2.603
180.0	373.0	345.4	1195.7	850.3	0.3945	2.53 5
185.0	375.23	347.8	1196.4	848.6	0.4049	2.47 0
190.0	377.4	350.1	1197.1	847.0	0.4153	2.40 8
195.0	379.6	352.4	1197.7	845.3	0.4257	2.34 9
200.0	381.7	354.6	1198.4	843.8	0.4359	2.29 4
250.0	401.0	374.7	1204.2	829.5	0.5393	1 85 4
300.0	417.4	391.9	1209.3	817.4	0.6440	1.55 4
400.0	444.9	419.8	1217.7	797.9	0.8572	1.16 7

TABLE IX. Weight of Water.

Temp. Fahr.	Heat Units per pound	Wt. lbs. per cu. ft.	Temp. Fahr.	Heat Units per pound	Wt. lbs. per cu. ft.	Temp. Fahr.	Heat Units per pound	Wt. lbs. per cu. ft.
32°	0.00	62.42	123	91.09	61.68	169	137.46	60.79
35	3.02	62.42	124	92.10	61.67	170	138.46	60.77
40	8.06	62.42	125	93.10	61.65	171	139.47	60.75
45	13.08	62.42	126	94.11	61.63	172	140.48	60.73
50	18.10	62.41	127	95.12	61.61	173	141.49	60.70
52	20.11	62.40	128	96.13	61.60	174	142.50	60.68
54	22.11	62.40	129	97.14	61.58	175	143.50	60.66
56	24.11	62.39	130	98.14	61.56	176	144.51	60.64
58	26.12	62.38	131	99.15	61.54	177	145.52	60.62
60	28.12	62.37	132	100.16	61.52	178	146.53	60.59
62	30.12	62.36	133	101.17	61.51	179	147.54	60.57
64	32.12	62.35	134	102.18	61.49	180	148.54	60.55
66	34.12	62.34	135	103.18	61.47	181	149.55	60.53
68	36.12	62.33	136	104.19	61.45	182	150.56	60.50
70	38.11	62.31	137	105.20	61.43	183	151.57	60.48
72	40.11	62.30	138	106.21	61.41	184	152.53	60.46
74	42.11	62.28	139	107.22	61.39	185	153.58	60.44
76	44.11	62.27	140	108.22	61.37	186	154.59	60.41
78	46.10	62.25	141	109.23	61.36	187	155.60	60.39
80	48.09	62.23	142	110.24	61.34	188	156.61	60.37
82	50.08	62.21	143	111.25	61.32	189	157.62	60.34
84	52.07	62.19	144	112.26	61.30	190	158.62	60.32
86	54.06	62.17	145	113.26	61.28	191	159.63	60.29
88	56.05	62.15	146	114.27	61.26	192	160.63	60.27
90	58.04	62.13	147	115.28	61.24	193	161.64	60.25
92	60.03	62.11	148	116.29	61.22	194	162.65	60.22
94	62.02	62.10	149	117.30	61.20	195	163.66	60.20
96	64.01	62.07	150	118.30	61.18	196	164.66	60.17
98	66.01	62.05	151	119.31	61.16	197	165.67	60.15
100	68.01	62.02	152	120.32	61.14	198	166.68	60.12
102	70.00	62.00	153	121.33	61.12	199	167.69	60.10
104	72.00	61.97	154	122.34	61.10	200	168.70	60.07
106	74.00	61.95	155	123.34	91.08	201	169.70	60.05
108	76.00	61.92	156	124.35	61.06	202	170.71	60.02
110	78.00	61.89	157	125.36	61.04	203	171.72	60.00
112	80.00	61.86	158	126.37	61.02	204	172.73	59.97
113	81.01	61.84	159	127.38	61.00	205	173.74	59.95
114	82.02	61.83	160	128.38	60.98	206	174.74	59.92
115	83.02	61.82	161	129.39	60.96	207	175.75	59.89
116	84.03	61.80	162	130.40	60.94	208	176.76	59.87
117	85.04	61.78	163	131.41	60.92	209	177.77	59.84
118	86.05	61.77	164	132.42	60.90	210	178.78	59.82
119	87.06	61.75	165	133.42	60.87	211	179.78	59.79
120	88.06	61.74	166	134.43	60.85	212	180.79	59.76
121	89.07	61.72	167	135.44	60.83			
122	90.08	61.70	168	136.45	60.81			

TABLE X.

The Properties of Anhydrous Ammonia.

Temperature Degrees F. t.	Pressure, p. (Absolute) Lbs. per sq. in.	Heat of Vaporization Thermal Units L.	External Heat $\frac{pu}{778}$ Thermal Units	Volume of Vapor per lb., cu. ft. v.	Volume of Liquid per lb., cu. ft. v_2
— 40	10.69	579.67	48.25	24.38	.0234
— 35	12.31	576.69	48.35	21.21	.0236
— 30	14.13	573.69	48.85	18.67	.0237
— 25	16.17	570.68	49.16	16.42	.0238
— 20	18.45	567.67	49.44	14.48	.0240
— 15	20.99	564.64	49.74	12.81	.0242
— 10	23.77	561.61	50.05	11.36	.0243
— 5	27.57	558.56	50.44	9.89	.0244
± 0	30.37	555.50	51.38	9.14	.0246
+ 5	34.17	552.43	50.84	8.04	.0247
+ 10	38.55	549.35	51.13	7.20	.0249
+ 15	42.93	546.26	51.33	6.46	.0250
+ 20	47.95	543.15	51.65	5.82	.0252
+ 25	53.43	540.03	51.81	5.24	.0253
+ 30	59.41	536.92	52.02	4.73	.0254
+ 35	65.93	533.78	52.22	4.28	.0256
+ 40	73.00	530.63	52,42	3.88	.0257
+ 45	80.66	527.47	52.62	3.53	.0260
+ 50	88.96	524.30	52.82	3.21	.02601
+ 55	97.93	521.12	53.01	2.93	.02603
+ 60	107.60	517.93	53.21	2.67	.0265
+ 65	118.03	515.33	53.40	2.45	.0266
+ 70	129.21	511.52	53.67	2.24	.0268
+ 75	141.25	508.29	53.76	2.05	.0270
+ 80	154.11	504.66	53.96	1.89	.0272
+ 85	167.86	501.81	54.15	1.74	.0273
+ 90	182.80	498.11	54.28	1.61	.0274
+ 95	198.37	495.29	54.41	1.48	.0276
+ 100	215.14	491.50	54.54	1.36	.0277